THE
INTERNAL-COMBUSTION
ENGINE

THE INTERNAL-COMBUSTION ENGINE

Volume II
HIGH-SPEED ENGINES

BY

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PREFACE

Slow-speed engines were described in the first volume of this work. In the present volume the author deals with high-speed engines.

He wishes again to express his indebtedness to his colleague, Mr. H. A. Hetherington, for his generous help, and to his assistants, Mr. J. F. Alcock, for his help and timely criticisms, and Mr. R. J. Cousins, for much valuable data, also to Mr. H. T. Tizard, for his friendly advice, and from whose splendid research work he has drawn very freely. His thanks are due also to the Asiatic Petroleum Company for their permission to publish freely and unreservedly the results of investigations carried out on their behalf into the behaviour of the various available liquid fuels; to the Vauxhall Motor Company for permission to describe not only one of their ordinary car engines, but also one of their special racing engines; to Mr. J. W. Burt for his kindness in preparing an analysis of sleeve valve operation; and to all those engine builders who have so kindly supplied him with data, etc.; and finally, but not least, to the Technical Department of the Air Ministry, to whose initiative so much research work on the Internal-Combustion Engine has been due.

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THE INTERNAL-COMBUSTION ENGINE—II

INTRODUCTION

Until the advent of the Great War, most of the scientific talent in this country which had interested itself in the development of the internal-combustion engine was devoted to research upon, and development of, the relatively heavy slow-speed stationary type, a type which, it now appears, has but a limited scope, for in the really large powers it can hardly compete with the modern steam turbine, while in the smaller powers, its field is being narrowed daily by the gradual spread of electricity.

Very soon after hostilities began, it became apparent that the light mobile high-speed type of internal-combustion engine applied to transport, aircraft, and later to tanks, was destined to play a very important, if not decisive, part in the conduct of the War. Every effort was then made to concentrate all the available scientific talent on the development of the high-speed engine. Independent scientists and investigators, and such national institutions as the Royal Aircraft Establishment and the National Physical Laboratory were requested to turn their attention to this subject, and every facility was lavished on them. They were invited to co-operate with the manufacturers and were asked to make a careful theoretical study of both the mechanical and thermodynamic problems involved, and to recommend how and in what direction the general efficiency of these light engines might be maintained and improved. The campaign of intensive research which resulted from this sudden influx of scientific talent, accompanied by almost inexhaustible funds for research, has resulted in the production of light high-speed engines which, besides giving what a few years ago would have been considered an almost incredible power

output in relation to their size and weight, can show as high an efficiency as that of the largest slow-speed type. What is perhaps more important still, the basic principles, both mechanical and thermo-dynamic, upon which the performance of such an engine depends, have been investigated in so complete and comprehensive a manner that the performance of any engine can now be gauged with accuracy from a study of the design alone; or conversely, an engine can be designed to fulfil any specific requirement as to power output or efficiency, with the same precision as in the case of a steam-engine.

That the internal-combustion engine has found its ultimate sphere in the light mobile high-speed type is now evidenced by the fact that, whereas in the years immediately before the War the annual output in horse-power of both the light and heavy type in this country was about equal, to-day the aggregate annual power output of the light high-speed type is at least ten times that of all other types, and in numbers probably nearer twenty times.

To-day far more is known about both the possibilities and the limitations of the high-speed internal-combustion engine than was the case before the War, and it seems fairly evident that its rôle lies in the propulsion of all forms of transport where its light weight and low fuel consumption render it supremely valuable. It is in the author's opinion extremely doubtful whether it will ever attain to more than its present very uncertain footing for stationary purposes, where neither its own light weight nor that of the fuel it consumes can be of much assistance to it in the struggle for existence.

Already, and in an incredibly short space of time, the internal-combustion engine has gained practically undisputed sway over all forms of road transport, and in doing so has developed and even almost revolutionized this previously decaying system. In a few short years it has both opened up the possibility of aerial transport and made it a powerful factor, in war at all events; it will almost certainly extend to rail transport also, beginning, no doubt, in countries where long distances have to be traversed and where fuel and water are scarce. It is already ousting the steam-engine from the smaller classes of shipping and is extending very gradually to the larger vessels, but here its progress will probably be slow, for the steam-turbine shows to particular advantage as a marine engine, because in this field alone it can always get that upon which its efficiency so largely depends, namely, an unlimited supply of cold water. Also the large steamship, alone of all forms of transport, requires a very

high-powered installation, and it is in units of high power output that steam retains its supremacy.

With but few exceptions, all light mobile engines consume volatile liquid fuel.

Up to the present the only two fuels available in bulk are petrol —a generic term covering any low-boiling distillate from crude petroleum—and benzol, a distillate from coal-tar, consisting of benzene with a small percentage of toluene.

The supply of both these fuels, and more particularly the latter, will soon become unequal to the demand, with the result that a critical situation is bound to arise in the not very distant future. Civilization is now so deeply committed to the use of internal-combustion engines for all road transport and for many other purposes, that it is a matter of absolute necessity to find an alternative fuel. Fortunately such a fuel is in sight in the form of alcohol; this is a vegetable product whose consumption involves no drain on the world's storage and which, in tropical countries at all events, can ultimately be produced in quantities, sufficient to meet the world's demand, at all events at the present rate of consumption.

By the use of a fuel derived from vegetation, mankind is adapting the sun's heat to the development of motive power, as it becomes available from day to day; by using mineral fuels, he is consuming a legacy—and a limited legacy at that—of heat stored away many thousands of years ago. In the one case he is, as it were, living within his income, in the other he is squandering his capital.

The mobile internal-combustion engine is now no longer a luxury; it has become one of the prime necessities of peaceful civilization and the prime necessity in time of war; therefore, the assurance of its fuel supply should be considered a matter of national importance. It is perfectly well known that alcohol is an excellent fuel, and there is little doubt but that sufficient supplies could be produced within the tropical regions of the British Empire, yet little or nothing is being done to encourage its development. Judging from past experience, no active steps will be taken until a serious crisis has arisen, and since it must take at least ten years to create the necessary organization and machinery for the production of alcohol on the scale which will be required, the crisis may be a serious and prolonged one.

In the author's opinion it is unlikely that any crisis in the fuel situation will have an adverse effect on the development of the

internal-combustion engine, for the simple reason that it has become a necessity, but it will probably have the effect of increasing the cost of transport and with it the cost of living generally. It will also, of course, have the effect of forcing designers of engines to concentrate more attention on the attainment of high efficiency and fuel economy, which is all to the good.

CHAPTER I

VOLATILE LIQUID FUEL FOR INTERNAL-COMBUSTION ENGINES

The volatile liquid fuels available in bulk at the present day, or likely to be available in the near future, consist of petrol, benzol, kerosene, and alcohol.

Petrol, as is well known, is a distillate from crude petroleum; it consists of a heterogeneous mixture of all those hydrocarbon fractions which boil between the limits of 140° F. and about 400° F.

These fractions belong to three different series:

	General Formula.
The Paraffins	C_nH_{2n+2}
The Naphthenes	C_nH_{2n}
and the Aromatics	C_nH_{2n-6}

In addition to these three leading groups there are also present a small proportion of members of the olefine series, though the proportion found in "natural" as opposed to "cracked" petrols is usually so small as to be almost insignificant.

The individual members of the paraffin series present in petrol are:

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F
Hexane	C_6H_{14}	156	0·663
Heptane	C_7H_{16}	209	0·691
Octane	C_8H_{18}	258	0·709
Nonane	C_9H_{20}	302	0·723
Decane	$C_{10}H_{22}$	343	0·735
Undecane	$C_{11}H_{24}$	383	0·746

Those of the naphthene series are:

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Cyclohexane	C_6H_{12}	178	0·780
Hexahydrotoluene	C_7H_{14}	212	0·770
Hexahydroxylene	C_8H_{16}	246·2	0·756

and those of the aromatic series:

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Benzene	C_6H_6	176	0·884
Toluene	C_7H_8	230	0·870
Xylene	C_8H_{10}	284	0·862

Although, in most examples of commercial petrol, the members of the paraffin series predominate, yet this is by no means always the case, and it is the exception rather than the rule, for the paraffin content of a petrol to exceed 60 per cent of the whole. Generally speaking, paraffins are commonest in the Western oilfields, naphthenes in the near Eastern, and aromatics in the far Eastern oilfields. This, however, is only a rough generalization, for there are many exceptions.

The following table gives the analysis of seven typical samples of petrol drawn from widely different parts of the world and illustrates how greatly the composition may vary. In this table the presence of small traces of other complex substances such as thiophine, &c., which play no perceptible part in the behaviour of the fuel, has been ignored.

Sample.	Approximate Composition by Weight per cent.			Specific Gravity.
	Paraffins.	Naphthenes.	Aromatics.	
A	26·0	35·0	39·0	0·782
B	62·0	23·0	15·0	0·723
C	61·0	30·5	8·5	0·727
D	38·0	47·0	15·0	0·760
E	68·0	20·0	12·0	0·719
F	80·0	15·2	4·8	0·704
H	10·0	85·0	5·0	0·767
Average	49·3	36·5	14·2	0·740

It is evident from the above that the specific gravity as a measure either of the composition or of the volatility of a fuel is quite meaningless. If, as is sometimes erroneously supposed, petrol consisted entirely of members of the paraffin series, then, since the gravity, the molecular weight, and the boiling point of the members of this series rise together, the gravity would be a measure of the volatility. The presence, however, of even a very small proportion of aromatics whose specific gravity ranges from 0·860 to about 0·885 will, of course, entirely upset any deductions which could be drawn from considerations of specific gravity alone. As a specific illustration it may be mentioned that of the fuels tabulated above, sample B sp. gr. 0·723, a special very low-boiling aircraft spirit prepared for the Cross-Atlantic flight, is by far the most volatile, yet its gravity is not the lowest by any means. The best fuels, i.e. those from which the highest power and efficiency could be obtained, were samples A, D, and H, which have the highest specific gravity of all, but which are rich in aromatics or naphthenes or both, while the worst without question is sample E sp. gr. 0·719.

It will be shown later that, of the three leading groups, the presence of the aromatics is the most of all to be desired from every point of view, that of the naphthenes next, while the paraffins are highly objectionable and the smaller the proportion present the better.

Though the phenomenon of detonation will be discussed later it may be stated at this stage that it is by far the most important factor in determining the quality of a fuel, and it is one which depends primarily upon its chemical composition. The paraffin series are the worst of all from this point of view, and they become progressively worse as their molecular weight and gravity increase; thus, for example, hexane is much better than heptane, and so on. The naphthenes are very much better, while the aromatics are the best of all as regards detonation.

Commercial benzol is a coal-tar distillate consisting primarily of pure benzene C_6H_6, with a little toluene and a trace of xylene. These are all aromatics. Its specific gravity ranges from 0·875 to 0·882, depending on the proportion of toluene present.

This fuel has many advantages over petrol, but fully to realize these, it is necessary to work with a much higher compression ratio.

The available members of the alcohol group consist of methyl, ethyl, and butyl alcohol. These are not true hydrocarbons, since

each contains oxygen in the molecule. In consequence of this and of their higher latent heat of evaporation, the behaviour of these fuels is somewhat different to that of the true hydrocarbons. So far as their tendency to detonate is concerned they are even better than the aromatics, though when used under a very high compression methyl alcohol in particular is liable to pre-ignite without warning. Owing to their high latent heat and low flame temperature the whole temperature of the cycle is lower, while owing again to their high latent heat of evaporation and therefore to the reduced suction temperature, the volumetric efficiency of an engine using alcohol is considerably higher than when using petrol or benzol. The result of the lower flame temperature is that the engine operates at a higher thermal efficiency, while the increase in volumetric efficiency much more than balances their lower internal energy, so that the maximum power output on alcohol is considerably greater, while the heat flow to the engine cylinder is lower than on petrol or benzol.

The properties of a fuel which determine its value for use in an internal-combustion engine are:

(i.) Tendency to detonate.
(ii.) Latent heat.
(iii.) Volatility.
(iv.) Calorific value of the fuel.
(v.) Heat value of the mixture.

All volatile liquid fuels when vaporized and mixed with air in the proportion required to give complete combustion have, within very close limits, the same heat value per standard cubic inch of mixture, hence they all give the same power and the same thermal efficiency when used under the same conditions. It is only by varying the compression ratio or by altering the degree of vaporization in the carburettor or induction pipe that any variation in power output or efficiency can be obtained.

Tendency of Fuels to detonate.—The phenomena of detonation as apart from its relation to the nature of the fuel will be dealt with later, but for the present it is sufficient to state that the limit to which the compression ratio can be raised, and therefore the limit of power output and efficiency, is governed by the conditions which control detonation and pre-ignition.

With all known petroleum spirits, detonation precedes and subsequently produces pre-ignition, but in the case of certain fuels such

as ether, carbon disulphide (at a low compression ratio), also the pure aromatics and alcohol (at a high compression ratio), pre-ignition is liable to occur without preliminary detonation.

Of the constituents of petrol the paraffins are by far the worst offenders as regards detonation, while the aromatics are the best. It is found also that all mixtures of these bodies obey the ordinary proportional laws and that there is practically a straight line relation between the mixture proportion of two or more hydrocarbons and the compression ratio at which detonation occurs.

In fig. 1 is shown as a full line the observed relation between

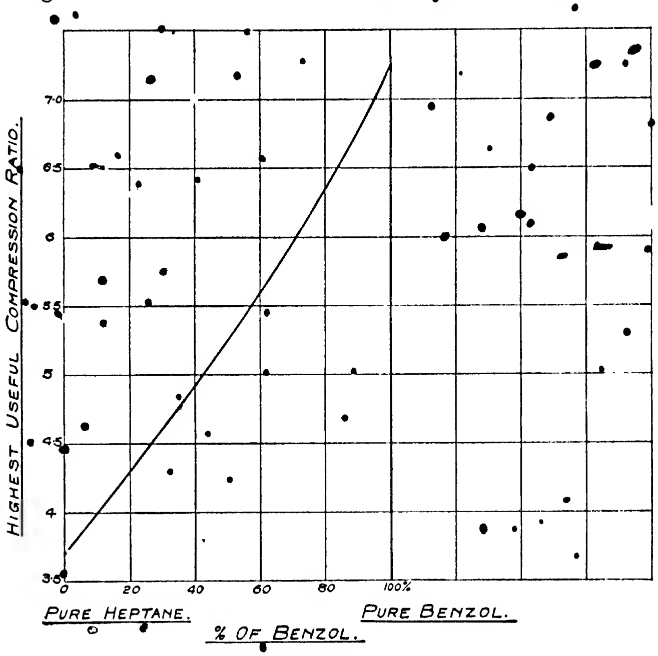


Fig. 1.—Curve showing Compression Ratio at which Detonation occurs

the compression ratio and the point at which detonation occurs when to pure heptane varying proportions of benzene (benzol) are added. Owing to the influence of combustion-chamber design, and to other factors which will be considered later, it is not possible to lay down a hard and fast relation between the fuel and the highest compression ratio at which it may be used in any type of engine, but it is possible in the light of present knowledge to give relative values, though there is some difficulty in selecting substances to be taken as standards. In the investigations which the author's firm carried

out on behalf of the Asiatic Petroleum Co., a sample of petrol, consisting mainly of paraffins from which nearly all the aromatics had been removed by sulphonation, was taken as zero, and at the other end of the scale pure toluene was chosen; the relative tendency of different fuels to detonate was then expressed in terms of their "toluene value," i.e. the equivalent proportion of toluene which it would be necessary to mix with the standard aromatic free petrol in order to give it the same tendency to detonate as that of the sample under examination. Later investigation showed that the standard "aromatic free," petrol which contained about 35 per cent of naphthenes and the lighter members of the paraffin series, was by no means the worst offender as regards detonation, and that in fact several samples of commercial petrol were actually considerably worse. Also it was found that toluene was not so effective in resisting detonation as ethyl alcohol. Since, however, the expression "toluene value" has become rather widely used, it is probably better to retain the term.

Table I gives the toluene values and the highest useful compression for various fuels. The highest useful compression ratio may be defined as the highest ratio at which a particularly efficient engine used for the purpose of investigating the behaviour of fuels could be operated without detonation at any mixture strength or with any ignition timing, with a standard amount of preheating to the carburettor, and at a speed of 1500 R.P.M.

In this connection reference may be made to the common belief that the rate of burning of the fuel, though one of the factors controlling detonation, forms a limit to the speed at which an engine can run.

The normal rate of burning (as distinct from the detonation rate) of any stagnant fuel/air mixture is so low as to be practically useless so far as any internal-combustion engine is concerned. We must look, therefore, entirely to turbulence or the mechanical distribution of the flame to spread combustion throughout the whole mass of the working fluid, and, since this is the case, it follows that the normal rate of burning of any fuel is practically without influence on the speed at which an engine will run.

It has been found that a fuel with a low normal rate of burning, such, for example, as ethyl alcohol, will operate just as efficiently in a high-speed, low-compression, engine as will hexane or petrol, and that the relative efficiency and power obtained is exactly the same throughout the whole speed range. The normal rate of burning of

TABLE I

Fuel.	Highest Useful Compression Ratio in variable Compression Engine. $\frac{\text{Total Volume}}{\text{Clearance Volume}}$	Toluene Value. Toluene = 100 %, Aromatic Free Petrol = 0 %.
Aromatic Free Petrol	4·85	0
" A " Petrol	6·0	38·0
" B " "	5·7	28·0
" C " "	5·25	13·5
" D " "	5·35	16·5
" E " "	4·7	– 5·0
" F " "	5·05	6·5
" G " "	4·55	– 10·0
" H " "	5·9	35·0
" I " "	4·3	– 20·0
Heavy Fuels		
Heavy Aromatics	6·5	55·0
Kerosene	4·2	– 22·0
Paraffin Series		
Pentane (Normal)	5·85	33·0
Hexane (80 % pure)...	5·1	8·0
Heptane (97 % pure)	3·75	– 37·0
Aromatic Series		
Benzene (pure)	6·9 †	67·0
Toluene (99 % pure)...	>7·0	100·0
Xylene (91 % pure)	>7·0	85·0
Naphthene Series		
Cyclohexane (93 % pure)	5·9 †	35·0
Hexahydrotoluene (80 %)	5·8	31·5
Hexahydroxylene (60 %)	4·9	1·5
Olefines		
Cracked Spirit (53 % unsat.) ...	5·55	23·5
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	>7·5	>88·0
" " (95 vol. %)	>7·5	>88·0
Methyl Alcohol (Wood Naphtha) ...	5·2 †	..
Methylated Spirits	6·5 †	..
Butyl Alcohol (Coml.)	7·3	80·0
Ether (50 % in petrol)	3·9	(– 32·0)
Carbon Disulph. (50 %)	5·15 †	(9·0)

NOTE.—This sign (†) indicates that pre-ignition occurred before audible detonation.

a fuel has, therefore, no connection with the speed at which an engine may be run.

In fact, since a slow-burning fuel is generally not prone to detonation, it is much more desirable than one which burns more rapidly.

Latent Heat.—The influence of the latent heat of evaporation of the fuel is a very important factor, but one which is usually ignored. This factor, coupled with the mean volatility, determines the density of the charge taken into the cylinder. It is, of course, clear that the weight of charge taken into the cylinder will, in any given case, be inversely proportional to its absolute temperature at the moment when the inlet valve closes. There is definite evidence from experimental results that, with the exception of alcohol and the other members of its group, all fuels boiling below about 400° F. are completely evaporated before the commencement of the compression stroke by contact with the hot walls and by admixture with the highly heated residual exhaust products in the cylinder; excepting only a very small proportion which may enter the cylinder in coarse drops, and so not only escape evaporation, but, even to a large extent, combustion also. This proportion is, however, quite insignificant, and has no influence, in so far as power output is concerned.

The absolute temperature at the commencement of the compression stroke is dependent upon (*a*) the amount of external heating applied, and (*b*) the latent heat of evaporation. It is largely independent of the temperature of the mixture during its entry to the cylinder. In fact, the final absolute temperature, and, therefore, the weight of the charge taken into the cylinder, are dependent upon the quantity and latent heat of the fuel, and upon the amount of heat added to it, external to the cylinder.

For example, a highly volatile fuel entering the cylinder at 40° F., and a fuel of low-vapour tension entering at 80° F., will both have the same final absolute temperature at the commencement of compression, if the latent heat of both is the same, and if both receive the same amount of pre-heating. In the former case most of the evaporation has taken place outside the cylinder, and the added heat has been absorbed by the latent heat of evaporation; in the latter case, little or no evaporation has taken place outside the cylinder, and the added heat has therefore raised the temperature of the air and of the, still liquid, fuel. In both cases contact and admixture with the highly heated exhaust products in the cylinder

will complete evaporation, and in both cases the final temperature will be the same, hence the weight of working fluid (which is inversely proportional to the absolute temperature at the end of the suction stroke), and therefore the power output, will be the same in both cases.

From the above considerations it will be seen that, with any given amount of pre-heating (provided it is not excessive), the volumetric efficiency, and therefore the power output, will increase with the latent heat of the fuel. Some calculations illustrating this point more fully will be found in Chapter II.

The above conclusions may be summed up as follows:—

(1) The power output is inversely proportional to the absolute temperature of the working fluid at the end of the suction stroke—since it is the temperature at this point which controls the weight of charge, and therefore the volumetric efficiency.

(2) Other things being equal, the final suction temperature is controlled by the amount of external heating, on the one hand, and the latent heat of the fuel, on the other; it is nearly independent of the temperature of the entering charge and of its volatility.

For any given fuel, therefore, the power output obtainable is controlled primarily by the amount of heat added to the working fluid before it enters the cylinder, and, so far as power output is concerned, it is immaterial whether the heat so added is devoted to raising the temperature of the mixture or to evaporating the fuel at low temperature. Conversely, with any given amount of pre-heating, the power output obtainable is dependent upon the latent heat of evaporation of the fuel.

Apart from the alcohol group, the variation in latent heat is not very large, and does not exercise any important influence. It is interesting to note, however, that in cases where the total internal energy is lower, the latent heat is generally slightly higher; consequently a slightly greater weight of charge is taken into the cylinder, sufficient in most cases to compensate for the lower internal energy, and thus bring the actual power output to substantially the same in all cases. This point is well illustrated by the instance of benzene, as will be seen later, or by reference to Table II. The energy liberated by the combustion of a cubic inch (at standard temperature and pressure) of benzene-air mixture is appreciably lower than that of the hydrocarbons forming the greater proportion of petrols. On the other hand, the latent heat of benzene is considerably greater, and as a result the power output obtainable under

TABLE II

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Total Energy liberated by Combustion, Ft. lb. per standard cub. in.	Relative Power, Output allowing for Increase in Density due to Evaporation. Octane = 100.
Paraffin Series			
Hexane	156	48·33	100·2
Heptane	133	48·64	100·1
Octane	128	48·73	100·0
Nonane	..	48·78	..
Decane	108	48·82	99·4
Aromatic Series			
Benzene	172	47·51	100·1
Toluene	151	47·98	100·0
Xylene	145	48·26	100·6
Naphthene Series			
Cyclohexane	156	48·11	100·0
Hexahydrotoluene ...	138	48·32	99·8
Hexahydroxylene ...	133	48·49	99·8

similar conditions from benzene is the same as that from petrol to within less than one half of 1 per cent.

The following Table III gives the latent heat of evaporation of a number of hydrocarbons and other substances enumerated in the previous tables. The air-to-fuel ratio by weight, also the drop in temperature of the mixture due to evaporation of the liquid, are shown for each fuel. The calculations are made for mixtures giving complete combustion, but without excess of air.

In the case of alcohol, owing to the very much higher latent heat and to the fact that the proportion of fuel to air is also much greater, the latent heat of evaporation plays a supremely important part, and results in a really marked increase in power as compared with other fuels, although the total internal energy of unit mass of mixture is lower than that of either petrol or benzol. Moreover, there is introduced a feature which is not observed to any marked extent with other fuels—namely, that the power output increases very considerably when an over-rich mixture is used, because more fuel is then evaporated, the temperature of the charge is lowered, and the gain in weight of charge considerably more than outweighs the loss due to the greater specific heat of the products of combustion.

In fig. 2 are shown the actual measured volumetric efficiency as

TABLE III

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Air-to-fuel Ratio (by Weight) for just complete Combustion.	Fall in Temperature of Mixture due to Latent Heat of Evaporation, ° F.
Paraffin Series			
Hexane...	156	15·2	37·8
Heptane	133	15·1	32·4
Octane	128	15·05	29·0
Nonane...	..	15·0	..
Decane	108	15·0	20·1
Aromatic Series			
Benzene	172	13·2	46·8
Toluene...	151	13·4	40·5
Xylene	145	13·6	38·7
Naphthene Series			
Cyclohexane	156	14·7	38·7
Hexahydrotoluene ...	138	14·7	34·2
Hexahydroxylene ...	133	14·7	32·4
Olefine Series			
Heptylene	167 (app.)	14·7	41·4
Decylene	..	14·7	..
Alcoholic Group			
Ethyl Alcohol	397	8·95	148·8
Methyl Alcohol ...	512	6·44	252·0
Miscellaneous			
Ether	158	11·14	49·5
Carbon Disulph. ...	153	9·35	55·8
Acetylene	Gas	13·2	..
Carbon Monox. ...	Gas	2·45	..
Hydrogen	Gas	34·3	..

The last column is calculated on the assumption that the specific heat of the fuel vapour is constant for all at 0·5.

found in the author's fuel research engine when using petrol and ethyl alcohol under precisely similar conditions as to temperature, &c., and at a compression ratio of 5 : 1. In both cases a careful series of measurements was made at mixture strengths ranging from 20 per cent weak to 25 per cent over-rich.

In the case of fuels whose volatility is very low, such as kerosene, butyl alcohol, &c., advantage cannot be taken of the latent heat of evaporation, because it then becomes necessary to add an excessive

amount of heat before entry to the cylinder, in order to prevent condensation in the induction system. For this reason alone the power output obtainable from kerosene is actually some 15 per cent lower than from petrol or other volatile hydrocarbons at the same compression ratio.

Volatility.—The mean volatility of a fuel is of importance since this determines the amount of pre-heating required to give reasonably uniform distribution. The amount of pre-heating governs, in its turn, the use which may be made of the latent heat of the liquid fuel. In single-cylinder engines volatility is, between wide limits, of

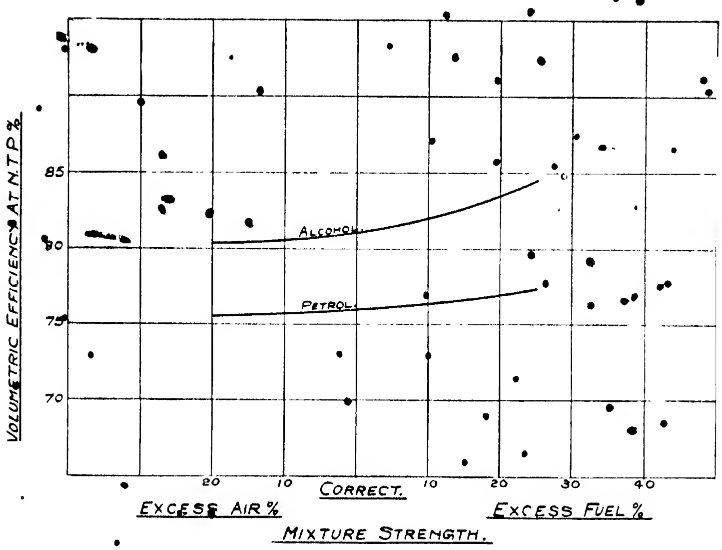


Fig. 2.—Observed Volumetric Efficiency on Petrol and Alcohol at different Mixture Strengths

comparatively little consequence since the exposed surface of the induction pipe is relatively small, but as the number of cylinders is multiplied and the length and surface of the induction system increased, so does the importance of volatility increase. A rough approximation (it is no more) of the relative volatility of different fuels can be obtained by measuring the rise or fall of temperature in the induction pipe of an engine, when a known weight of fuel and air are passing into it and when the amount of heat supplied to the carburettor can be recorded accurately.

The figures in the following Table IV give some clue as to the relative volatility of the different fuels—all were tested under exactly

TABLE IV

Fuel.	Rise or Fall of Temp. in Induction Pipe (indicating app. Mean Volatility). $\frac{\text{Heat 65 B.Th.U.}}{\text{Min.}}$ Deg. F.	Fall in Temperature (calculated) of $\frac{\text{Air}}{\text{Fuel}}$ Mixture due to Latent Heat of Evaporation. Deg. F.
Aromatic Free Petrol	+18	32·4
"A" Petrol	+19·8	36·0
"B" ,, ...	+5·4	34·2
"C" ,, ...	+12·6	33·3
"D" ,, ...	+19·8	33·0
"E" ,, ...	+18·9	32·8
"F" ,, ...	+9·5	32·8
"G" ,, ...	+32·4	...
"H" ,, ...	+21·6	36·0
"I" ,, ...	+25·2	...
Heavy Fuels		
Heavy Aromatics	+50·5	[illegible]
Kerosene	+56	26·1
Paraffin Series		
Pentane (Normal)	...	37·8
Hexane (80 % pure)...	0	37·8
Heptane (97 % pure)	+10	32·4
Aromatic Series		
Benzene (pure)	−13·1	16·8
Toluene (99 % pure)...	+14·4	10·5
Xylene (91 % pure)	+32·4	38·7
Naphthene Series		
Cyclohexane (93 % pure)	−5·4	38·7
Hexahydrotoluene (80 %)	+5·4	34·2
Hexahydroxylene (60 %)	+24·3	32·4
Olefines		
Cracked Spirit (53 % unsat.) ...	+19	37·5
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	+2·7	153·0
,, ,, ([illegible] vol. %)	−3·6	176·0
Methyl Alcohol (Wood Naphtha) ...	−15·3	252·0
Methylated Spirits	−1·8	198·0
Butyl Alcohol (Coml.)	+18	...
Ether (50 % in petrol)	+1·8	39·6
Carbon Disulph. (50 %)	−12·6	48·6

the same conditions as regards speed, temperature, heat input, &c. The temperature measured in the induction pipe and recorded in the table are, in each instance, those found when the mixture strength was that giving complete combustion.

The last column gives the calculated drop in temperature assuming no heat input and that the whole of the fuel were evaporated before entry to the cylinder.

Final Boiling Point.—It is always desirable to keep the final boiling point of any fuel as low as possible, because the higher boiling fractions are liable to condense on the cylinder walls and so to pass down into the crankcase, where they foul the lubricating oil.

As a general rule, so long as the final boiling point does not exceed say 400° F. no serious trouble need be feared on this score, for, if any fuel does condense on the walls, it will evaporate off in the crankcase, but in the case of kerosene and other high boiling fuels condensation on the cylinder walls and in the crankcase is one of the most serious difficulties with which designers of engines using these fuels have to contend.

Starting.—The readiness of a fuel to start from cold depends upon the proportion of low boiling-point fractions present in the fuel.

With nearly all commercial petrols the full vapour pressure at normal atmospheric temperature is only reached when at least 3 per cent of the volume of the vessel is occupied with liquid.

In an actual engine, starting is required with a minimum of about one-thirtieth of this, and even then the mixture strength would be about nine times richer than the normal running mixture. It follows, therefore, that unless a fuel has an abnormally high vapour tension it is necessary always to provide a very large excess for starting, e.g. by flooding or by the use of a special pilot jet, &c., as in the Zenith carburettor or by other means.

In the case of composite fuels such as petrol, the readiness to start depends rather upon the proportion of low boiling-point fractions present in the fuel than upon the *mean* boiling point. In the case of homogeneous fuels, even though of low boiling point, such as alcohol, starting from cold may be quite impossible, and it is then necessary to add some other fuel which either has itself, or by its admixture imparts, a much higher vapour tension. Ether may be taken as an example of the former and benzene or benzol of the latter.

Calorific Value.—The heat liberated by the combination of the

fuel and air is usually determined by burning the fuel in some form of calorimeter. The heat value so found includes the latent heat of the water formed, because in any form of calorimeter the temperature necessarily falls below the boiling point of water. Since, however, it is quite out of the question in any internal-combustion engine to make use of the latent heat of the water, it is customary to deduct from the total heat liberated that due to the condensation of the water formed. The heat value found after this deduction has been made, is termed the lower calorific value of the fuel and is generally accepted as the basis upon which to calculate the thermal efficiency of an engine. In the case of internal-combustion engines using volatile liquid fuels, such a value for the available heat is not quite correct because, when a fuel is burnt in a bomb or other calorimeter, some of the heat of combustion is devoted to evaporating the remainder of the liquid fuel and is therefore absorbed. Now when used in an engine the whole of the liquid is evaporated before combustion takes place, and the heat required for its evaporation is supplied by the waste heat from the cycle or by the available heat already present in the air. In either case it is supplied by heat other than the heat of combustion of the fuel. If, therefore, it be accepted as correct that the latent heat of evaporation of the water formed should be deducted from the total heat of combustion as determined by calorimetric measurement, because this heat cannot be utilized, then it is equally right and proper that the latent heat of evaporation of the liquid fuel itself should be added to the observed calorimetric determination because its equivalent value in the heat of combustion can be and is used in any internal-combustion engine in which the fuel is evaporated before combustion starts, i.e. in any but Diesel engines; strictly speaking, there should be added the latent heat at constant volume, which is less than that at constant pressure by an amount equal to the work done on the atmosphere if the liquid is evaporated when unenclosed. In the following Table V is given the heat of combustion of various fuels in terms of B.Th.U.s per lb. and per gallon, the first two columns being the usually accepted lower calorific value and the second the revised value corrected to include the latent heat of evaporation of the liquid fuel itself. This latter corrected value is used throughout this volume for all determinations of thermal efficiency.

It cannot be too strongly emphasized that the heat value of a volatile liquid fuel bears no relation whatever to the power output

TABLE V

Fuel	Calorific (lower) Value (exclusive of Latent Heat). B.Th.U. per lb.	Calorific (lower) Value (exclusive of Latent Heat). B.Th.U. per gallon.	Calorific (lower) Value (including Latent Heat at Constant Volume). B.Th.U. per lb.	Calorific (lower) Value (including Latent Heat at Constant Volume). B.Th.U. per gallon.	Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.
Aromatic Free Petrol	19,080	137,000	19,200	136,200	133·0
"A" Petrol	18,450	144,300	18,580	145,200	142·0
"B" "	18,890	136,600	19,020	137,500	140·0
"C" "	19,000	138,100	19,120	137,000	135·0
"D" "	18,770	142,600	18,890	143,500	132·0
"E" "	18,970	136,400	19,090	137,100	132·0
"F" "	19,130	134,700	19,250	135,500	134·0
"G" "	...	...	...	...	...
"H" "	18,790	144,100	18,920	145,000	145·0
"I" "	...	...	...	...	...
Heavy Fuels					
Heavy Aromatics	17,900 (App.)	158,500 (App.)	18,030	159,600	136·0
Kerosene	19,000 (App.)	154,400 (App.)	19,100	155,200	108·0
Paraffin Series					
Pentane (Normal)	19,600	122,300	19,740	123,100	154·0
Hexane (80 % pure)	19,250	131,900	19,390	132,900	156·0
Heptane (97 % pure)	19,300	132,800	19,420	134,100	133·0
Aromatic Series					
Benzene (pure)	17,302	152,950	17,460	154,200	172·0
Toluene (99 % pure)	17,522	152,500	17,660	153,600	151·0
Xylene (91 % pure)	17,800	153,500	17,930	154,500	145·0
Naphthene Series					
Cyclohexane (93 % pure) ...	18,800	147,800	18,940	149,000	156·0
Hexahydrotoluene (80 %) ...	18,760	146,200	18,880	147,200	138·0
Hexahydroxylene (60 %) ...	18,770 (App.)	139,700 (App.)	18,890	140,600	133·0
Olefines					
Cracked Spirit (53 % unsat.) ...	18,400 (App.)	139,400 (App.)	18,540	140,200	150·0 (App.)
Alcohol Group, &c.					
Ethyl Alcohol (98 %)	11,480	91,600	11,840	94,500	406·0
" " (95 vol. %) ...	10,790	88,000	11,130	92,000	442·0
Methyl Alcohol (Wood Naphtha)	9,630	79,900	10,030	83,300	500·0 (App.)
Methylated Spirits	10,260	83,700	10,580	86,900	450·0 (App.)
Butyl Alcohol (Coml.)	...	...	...	...	...
Ether (50 % in petrol)	16,700 (App.)	121,300 (App.)	16,830	122,500	146·0 (App.)
Carbon Disulph. (50 %) ...	10,600	105,400	10,730	106,600	146·0

obtainable from that fuel. In the case of gaseous fuels when the calorific value is very low, the apparent relationship is due largely to the presence in the gas of a considerable proportion of inert diluents such as nitrogen or carbon dioxide, in which case, of course, the power output is reduced because these diluents displace some of the available oxygen. In the case of volatile liquid fuels, however, the vapour contains no diluents at all, and the power output available is, therefore, entirely independent of the calorific value of the fuel.

The heat value of a fuel is, on the other hand, a direct measure of the quantity of fuel required, the lower the heat value the greater being the quantity needed to do the same work..

Heat Value of Mixture.—It is upon the heat value of the mixture of fuel and air, in the proportion required to give complete combustion, that the power output of an engine depends, and in this connection we find that all hydrocarbon fuels give within very close limits the same heat value per standard cubic inch of correct mixture. When allowance is made for the increase or decrease in specific volume after combustion, the variation becomes even less.

The following Table VI gives :—

Col. (1) The corrected calorific value of various petrols and other fuels.
Col. (2) The ratio of air/fuel by weight for complete combustion.
Col. (3) The increase or decrease in specific volume after combustion.
Col. (4) The ft.-lb. of energy liberated by the combustion of one standard cubic inch of mixture giving complete combustion, i.e. the total available internal energy.

The heat value of the " correct " mixture is usually termed the total " internal energy " of the working fluid, and this term will in future be used in order to distinguish it from the calorific value of the fuel, which latter has no influence on the power output.

Thermal Efficiency obtainable from Different Fuels.—Provided that the fuel is reasonably volatile, the thermal efficiency obtainable at any given compression ratio is substantially the same for all hydrocarbon fuels, irrespective of their chemical composition or of any other factor. In the case of the alcohol group, however, a somewhat higher thermal efficiency is obtained because, owing in part to their higher latent heat, and in part to their lower flame temperature, both the mean and the maximum temperatures of the cycle are lower, and the losses are therefore somewhat less. The range of burning on the weak side, which, by controlling the flame temperature, would control also the efficiency, happens to be almost

TABLE VI

Fuel.	(1) Calorific (lower) Value (including Latent Heat at Constant Volume). B.Th.U. per lb.	B.Th.U. per gall.	(2) Air-to-Fuel Ratio by Weight for Complete Combustion.	(3) Spec. Volume after Combustion — that before Combustion or "Volume Ratio."	(4) Total Energy liberated by Combustion per cub. in. at n.t.p. of Mixture giving complete Combustion foot-lb.
Aromatic Free Petrol	19,200	136,200	15·05	1·053	48·5
"A" Petrol	18,580	145,200	14·3	1·038	48·15
"B" "	19,020	137,500	14·7	1·049	48·45
"C" "	19,120	137,000	14·8	1·052	48·53
"D" "	18,890	143,500	14·6	1·047	48·35
"E" "	19,090	137,100	14·9	1·051	48·51
"F" "	19,250	135,500	15·0	1·053	48·54
"G" "	...	...	...	...	...
"H" "	18,920	145,000	14·7	1·048	48·31
"I" "	...	...	...	...	...
Heavy Fuels					
Heavy Aromatics	18,030	159,600	13·8	1·04	48·52
Kerosene	19,100	155,200	15·0	1·06	48·91
Paraffin Series					
Pentane (Normal)	19,740	123,100	15·25	1·051	48·7
Hexane (80 % pure)	19,390	132,900	15·2	1·051	48·35
Heptane (97 % pure)	19,420	134,100	15·1	1·056	48·64
Aromatic Series					
Benzene (pure)	17,460	154,200	13·2	1·013	47·51
Toluene (99 % pure)	17,660	153,600	13·4	1·023	47·98
Xylene (91 % pure)	17,930	154,500	13·6	1·03	48·1
Naphthene Series					
Cyclohexane (93 % pure) ...	18,940	149,000	14·7	1·044	48·11
Hexahydrotoluene (80 %) ...	18,890	147,200	14·7	1·047	48·2
Hexahydroxylene (60 %) ...	18,890	146,600	14·8	1·054	48·59
Olefines					
Cracked Spirit (53 % unsat.)...	18,540	140,200	14·8 (App.)	1·054	49·54
Alcohol Group, &c.					
Ethyl Alcohol (98·5 %) ...	11,840	94,500	8·9	1·065	47·39
" " (95 vol. %) ...	11,130	92,000	8·4	1·065	46·86
Methyl Alcohol (Wood Naphtha)	10,030	83,300	6·5	1·06 (App.)	48·2 (App.)
Methylated Spirit	10,580	86,900	8·0 (App.)	1·064	48·82
Butyl Alcohol (Coml.)	...	...	...	...	...
Ether (50 % in petrol)... ...	16,830	122,500	13·0	1·06	49·2
Carbon Disulph. (50 %) ...	10,130	106,600	10·8	0·98	39·4

exactly the same in the case of all volatile liquid fuels yet examined with the exception of ether, and, in all cases, the maximum thermal efficiency is obtained when the mixture is 15 per cent weak. If any attempt be made to weaken the mixture beyond this point, combustion becomes unduly slow and incomplete, and the efficiency falls away in consequence. Theoretically, of course, the efficiency should rise steadily as the mixture strength is reduced and should follow an almost straight line law until at the point when the mixture is infinitely weak the efficiency should be equal to the air standard. In practice the extent to which the mixture can be weakened with increase of efficiency depends to some small extent, in the case of a single-cylinder engine, upon the position of the sparking plug and the intensity of the spark, while in a multi-cylinder engine it depends to a much larger extent upon the uniformity of distribution, but in both cases it is very limited.

It is perhaps rather remarkable that the maximum thermal efficiency obtainable with two fuels so widely different as hexane C_6H_{14} and benzene C_6H_6 should be the same, but the explanation lies in the fact that while for CO_2 the dissociation is great at high temperatures, yet the increase in specific heat is small. On the other hand, for H_2O the dissociation is small but the increase in specific heat is great. Curiously enough, these effects almost exactly balance one another, so that the sum of the losses from each source is practically the same.

The following Table VII gives :—

Col. (1) The lower calorific value of the fuel corrected for the latent heat of evaporation.

Col. (2) The lowest fuel consumption in terms of lbs. and pints per I.H.P. hour at a compression ratio of 5 : 1.

Col. (3) The corresponding thermal efficiency.

The above figures relate to the thermal efficiency obtainable when all fuels are used at the same compression ratio.

It has, however, been stated previously that the highest compression ratio at which a fuel may be used depends upon its tendency to detonate, and it has been shown that this varies widely.

The following Table VIII shows the fuel consumption in terms both of pints and lbs. per I.H.P. hour when each fuel is used at its highest useful compression ratio.

It will be noted that in the case of kerosene and one or two other examples which are not very volatile, the observed thermal efficiency is considerably lower, due to the fact that a substantial

TABLE VII

Fuel.	(1) Calorific (lower) Value including Latent Heat at Constant Volume).		(2) Minimum Consumption at Compression Ratio of 5 : 1 per I.H.P. Hour.		(3) Thermal Efficiency at Compression Ratio of 5 : 1 per cent.
	B.Th.U. per lb.	B.Th.U. per gallon.	lb.	Pints.	
Aromatic Free Petrol	19,200	136,200	0·415*	0·462*	31·9*
"A" Petrol	18,580	145,200	0·432	0·442	31·7
"B" ,, ...	19,020	137,500	0·423	0·468	31·7
"C" ,, ...	19,120	137,000	0·421	0·463	31·6
"D" ,, ...	18,890	143,500	0·422	0·445	31·9
"E" ,, ...	19,090	137,100	0·421*	0·469*	31·7*
"F" ,, ...	19,250	135,500	0·414	0·471	31·9
"G" ,, ...	...	...	0·426	0·454	...
"H" ,, ...	18,920	145,000	0·425	0·443	31·7
"I" ,, ...	...	...	0·418	0·460	...
Heavy Fuels					
Heavy Aromatics	18,030	159,600	0·510	0·461	27·6
Kerosene	19,100	155,200	0·523*	0·515*	25·4*
Paraffin Series					
Pentane (Normal)	19,740	123,100	...	...	...
Hexane (80 % pure)	19,390	132,900	0.411	0·480	32·0
Heptane (97 % pure)	19,420	134,100	0·410*	0·475*	31·9*
Aromatic Series					
Benzene (pure)	17,460	154,200	0·458	0·415	31·8
Toluene (99 % pure)	17,660	153,600	0·455	0·418	31·7
Xylene (91 % pure)	17,930	154,500	0·452	0·420	31·4
Naphthene Series					
Cyclohexane (93 % pure) ...	18,940	149,000	0·420	0·427	31·9
Hexahydrotoluene (80 %) ...	18,890	147,200	0·425	0·430	31·7
Hexahydroxylene (60 %) ...	18,890	140,600	0·424*	0·456*	31·8*
Olefines					
Cracked Spirit (53 % unsat.) ...	18,540	140,200	0·429	0·453	32·0
Alcohol Group, &c.					
Ethyl Alcohol (98 %)	11,840	94,500	0·663	0·665	32·4
,, ,, (95 vol. %) ...	11,130	92,000	0·705	0·692	32·5
Methyl Alcohol (Wood Naphtha)	10,030	83,300	0·777	0·750	32·7
Methylated Spirits	10,580	86,900	0·740	0·721	32·5
Butyl Alcohol (Coml.)	...	...	0·566	0·550	
Ether (50 % in petrol)... ...	16,830	122,500	...	...	...
Carbon Disulph. (50 %) ...	10,730	106,600	...	...	...

* This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5 : 1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.

TABLE VIII

Fuel.	Minimum Consumption at Highest Useful Compression per I.H.P. Hour. lb.	Pints.
Aromatic Free Petrol	0·422	0·471
"A" Petrol	0·393	0·402
"B" "	0·393	0·435
"C" "	0·410	0·451
"D" "	0·407	0·428
"E" "	0·435	0·484
"F" "	0·412	0·469
"G" "	0·449	0·478
"H" "	0·389	0·405
"I" "	0·457	0·503
Heavy Fuels		
Heavy Aromatics	0·447	0·404
Kerosene	0·581	0·571
Paraffin Series		
Pentane (Normal)	..	..
Hexane (80 % pure)	0·405	0·473
Heptane (97 % pure)	0·491	0·568
Aromatic Series		
Benzene (pure)	0·392	0·355
Toluene (99 % pure)	0·385	0·354
Xylene (81 % pure)	0·381	0·354
Naphthene Series		
Cyclohexane (93 % pure)...	0·385	0·392
Hexahydrotoluene (80 %)	0·391	0·404
Hexahydroxylene (60 %)...	0·429	0·461
Olefines		
Cracked Spirit (53 % unsat.)	0·405	0·428
Alcohol Group, &c.		
Ethyl Alcohol (98 %)	0·532	0·533
" " (95 vol. %)	0·565	0·555
Methyl Alcohol (Wood Naphtha) ...	0·725	0·700
Methylated Spirits...	0·625	0·609
Butyl Alcohol (Coml.)	0·472	0·459
Ether (50 % in petrol)	..	..
Carbon Disulph. (50 %)	..	..

proportion of the liquid fuel is deposited on the walls of the induction piping and cylinder and escapes combustion. This proportion

could be reduced by further pre-heating, but all the tests were carried out, for comparative purposes, at exactly the same heat input to the carburettor.

By additional pre-heating a slightly higher thermal efficiency can be obtained, but the power output is reduced and the tendency to detonate increased thereby.

The Maximum Power Output.—The maximum power output obtainable from any fuel depends upon the internal energy of the working fluid and upon the latent heat of evaporation of the liquid.

The former varies very little, the latter considerably as between different fuels—generally speaking as regards the true hydrocarbon fuels the variations in internal energy and latent heat just about balance, with the result that the maximum power output is the same for all. For example, the total internal energy of benzene is about 1·5 per cent less than that of hexane, on the other hand the latent heat of benzene is considerably greater, and a greater weight of mixture is therefore retained in the cylinder, with the result that under identical temperature and other conditions both give, at the same compression ratio, the same power output to within less than half of 1 per cent.

In the case of alcohol, although the total internal energy of the mixture is appreciably lower, yet the latent heat is so much greater that a much denser charge is retained in the cylinder and the power output is some 5 per cent greater despite the lower internal energy.

Table IX shows:

Col. (1) The total internal energy.
Col. (2) The latent heat of evaporation.
Col. (3) The observed indicated mean pressure (power output) at a compression ratio of 5 : 1.
Col. (4) The observed indicated mean pressure at the highest useful compression ratio.

Fuels for Aircraft.—For all commercial purposes other than aircraft, fuel is supplied by bulk, not weight, and it is therefore the heat value per gallon and not per lb. which need be considered. In the particular case of aircraft, however, the weight and not the bulk of the fuel becomes the primary consideration. Other things being equal, therefore, the fuel with the highest calorific value per lb. will be the most efficient. From this point of view alone, the paraffin series would appear the most desirable. Unfortunately, however, owing to their tendency to detonate, the members of this series cannot be used in a high compression and therefore in an efficient engine.

TABLE IX

Fuel.	Total Energy liberated by Combustion per cub. in. of Mixture giving complete Combustion ft-lb.	Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.	Max. Ind. Mean Effective Pressure at Compression Ratio of 5 : 1. Heat 65 B.Th.U./Min. lb per sq. in	Max. Ind. Mean Effective Pressure at Highest Useful Compression. Heat 65 B.Th.U./Min. lb per sq. in.
Aromatic Free Petrol	48·5	133·0	131·3*	150·0
"A" Petrol	48·15	142·0	131·2	140·1
"B" ,, ...	48·45	140·0	131·5	137·5
"C" ,, ...	48·53	135·0	131·0	133·9
"D" ,, ...	48·35	132·0	131·2	131·9
"E" ,, ...	48·51	132·0	131·0*	128·6
"F" ,, ...	48·51	134·0	131·8	132·7
"G" ,, ...	..	..	131·5*	127·4*
"H" ,, ...	48·31	145·0	131·0	139·5
"I" ,, ...	..	..	131·7*	125·1
Heavy Fuels				
Heavy Aromatics	48·52	136·0	131·1	112·5
Kerosene...	48·91	108·0	130·7	123·0
Paraffin Series				
Pentane (Normal)	48·7	154·0	131·3	139·0
Hexane (80 % pure)	48·35	156·0	132·3	133·1
Heptane (97 % pure)	48·61	133·0	131·2*	119·5
Aromatic Series				
Benzene (pure)	47·51	172·0	131·6	116·5
Toluene (99 % pure)	47·98	151·0	131·5	147·0
Xylene (91 % pure)	48·1	145·0	131·5	146·8
Naphthene Series				
Cyclohexane (93 % pure) ...	48·11	156·0	131·3	139·0
Hexahydrotoluene (80 %) ...	48·2	138·0	131·0	137·9
Hexahydroxylene (60 %) ...	48·59	133·0	130·8*	130·0
Olefines				
Cracked Spirit (53 % unsat.) ...	49·54	150·0 (App.)	131·6	136·0
Alcohol Group, &c.				
Ethyl Alcohol (98 %)	47·39	406·0	137·8	156·5
,, ,, (95 vol. %) ...	46·86	442·0	142·0	161·2
Methyl Alcohol (Wood Naphtha)	48·2 (App.)	500·0 (App.)	144·8	146·6
Methylated Spirits	46·82	450·0 (App.)	144·5	155·5
Butyl Alcohol (Coml.)	..	..	138·0	156·0
Ether (50 % in petrol)... ...	49·2	146·0 (App.)	136·0*	125·0
Carbon Disulph. (50 %) ...	39·4	146·0	124·5	125·7

* This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5 : 1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.

If the compression were adjusted to suit the fuel, then the greatest distance could be flown on that fuel which gives the highest value for the product thermal efficiency × heat value per lb., and it is interesting to note that of all the fuels examined the highest figures are:

	Efficiency per cent.	Heat Value B.Th.U. per lb.	Product.
(1) Xylene	0·373 ×	17,930 =	6700
(2) Cyclohexane	0·349 ×	18,940 =	6600
(3) Petrol (sample B) ...	0·341 ×	19,020 =	6500

Although xylene comes first on the list, it is not a practicable fuel on account of its high boiling point and low volatility, also it is only efficient when used at a compression ratio so high (namely, above 6·75 : 1) as to render the engine unduly heavy owing to the very high maximum pressures to be withstood. Cyclohexane, one of the lighter members of the naphthene series, would actually give the best possible results from every point of view, but is not obtainable in bulk. The next on the list, viz. the petrol sample (B), is a very light and highly volatile fuel which was prepared by the Asiatic Petroleum Co. for, and used in, the cross-Atlantic flight.

In aircraft engines it is particularly desirable to use a very volatile fuel on account of distribution, and in order to be able to reduce to the minimum the amount of pre-heating required, and so to make the maximum possible use of the latent heat of evaporation to increase the power output.

For present-day aircraft engines which have an average compression ratio of about 5·4 : 1 the fuel used should have a toluene value of not less than +20, that is to say, it should contain not less than about 25 per cent of aromatics or their equivalent in naphthenes, in order to eliminate detonation and so enable the engine to develop its full power at ground level, as is most necessary when starting heavily laden. Since, however, detonation is largely a function of pressure, once an aeroplane has attained a reasonable altitude and is in air at a lower density, the tendency to detonate will disappear and a fuel of lower toluene value may be used. From an idealist point of view, therefore, it would seem desirable to employ a fuel of high toluene value at or near ground level, and to change over to a nearly pure paraffin petrol so soon as a sufficient altitude has been obtained.

By the use of alcohol, or of a fuel with high latent heat, or of one which has had its latent heat of evaporation augmented by the

introduction of water in solution with the fuel, a very substantial advantage may be gained, because, as shown previously, the high latent heat, will increase the available power output very considerably,—a most important consideration, especially in the case of aeroplanes when taking off with heavy loads.

SUMMARY

The results of a lengthy investigation of the various volatile liquid fuels carried out on behalf of the Asiatic Petroleum Co. may be summarized as follows :—

(1) It has been proved that the tendency of a fuel to detonate is the one outstanding factor in determining its value for use in a constant-volume internal-combustion engine. Compared with this, most other considerations are of secondary importance.

(2) There appears to be little doubt as to the correctness of the view, now generally accepted, that detonation is largely dependent upon the normal rate of burning of the fuel and is less the lower the rate of burning.

(3) In all cases it seems that a low rate of burning is advantageous. No fuel has yet been found whose rate of burning was too low to permit of maximum efficiency being obtained in the highest speed engine yet tested.

(4) Fuels capable of standing a very high compression will operate in a low compression engine, equally as efficiently as those whose normal rate of burning is high—provided that there is a reasonable degree of turbulence in the combustion chamber.

(5) Apart from the limitations introduced by detonation, the power output obtainable from all volatile liquid fuels, with the exception of the alcohol group, is the same at the same compression to within less than 2 per cent. Such variations as occur, within this range, are due rather to variations in the latent heat of evaporation than to any other circumstance.

(6) Owing to the high latent heat and low boiling point of alcohol and certain other bodies, the weight of charge per cycle is greater and a higher power output is obtained in consequence.

(7) The efficiency with which all volatile fuels, other than alcohol, &c., are burnt is practically the same, at the same compression ratio, irrespective of rate of burning, provided the compression is low enough to avoid detonation under any circumstances. In the case of alcohol, the efficiency is slightly higher, on account of the lower flame temperature.

(8) The useful range of burning is, to all intents and purposes, the same for all volatile liquid fuels.

(9) The unavoidable losses due to the combined influence of dissociation and change of specific heat at high temperatures are substantially the same in all cases.

(10) All the experimental results indicate that the performance of any combination of hydrocarbons as regards detonation, and therefore the power output and efficiency obtainable, is the mean performance of each of the components. The performance of any complex fuel such as petrol can therefore be predicted, once the nature and proportion of its constituents are determined; or conversely, a fuel can be prepared to give any required performance, within the limits available.

(11) The highest useful compression ratio for, and therefore the power output obtainable from, any petrol is governed by the relative proportions of aromatics, naphthenes, and paraffins it contains—the smaller the proportion of the latter the better from almost every point of view.

(12) To judge of the quality of a fuel by its specific gravity is entirely misleading. If naphthene and aromatic fractions are present in any large proportion (as is frequently the case), then a high specific gravity is a substantial advantage.

(13) Owing to the very rich mixture delivered normally by pilot jets, and the still further enrichment effected by flooding, only a relatively small proportion of highly volatile constituents appears to be required for starting.

Table X (facing p. 30) and Table XI (facing p. 32) give a general summary of the above investigations and test results which were carried out on behalf of the Asiatic Petroleum Co. That the author is permitted to publish them without reservation of any kind is due to the generosity and public spirit of this most enterprising company. The following particulars of the single-cylinder research engine and its accessories which formed the principal piece of apparatus used for the research, may be of some interest:—

In figs. 3–6 are shown drawings and photographs of the variable compression engine referred to in connection with the preceding tests.

In the design of this engine the following considerations were taken into account:—

(1) In view of the prolonged and extensive nature of the tests, not only were durability and reliability regarded as matters of

	G.		H.	I.	J.	K.	L.	M.
	lorific (lower) Value (including Latent Heat at Constant Volume)		Heat of Combustion per cubic inch at n t p of Mixture giving complete Combustion foot-lb	Spec Volume after Combustion what before Combustion or "Volume Ratio"	Total Energy liberated by Combustion per cub in at n t p of Mixture giving complete Combustion foot-lb	Air-to-Fuel Ratio by Weight for complete combustion	Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric). B.Th U. per lb	Fall in Temperature (calculated) of Air-Fuel Mixture due to Latent Heat of Evaporation. Deg F.
	B Th U. per lb	B Th U per gallon						
Aroma								
" A "	19,200	136,200	46·08	1·053	48·5	15·05	133·0	32·4
" B "	18,580	145,200	46·39	1·038	48·15	14·3	142·0	36·9
" C "	19,020	137,500	46·19	1·049	48·45	14·7	140·0	34·2
" D "	19,120	137,000	46·13	1·052	48·53	14·8	135·0	33·3
" E "	18,890	143,500	46·18	1·047	48·35	14·6	132·0	33·0
" F "	19,090	137,100	46·16	1·051	48·51	14·9	132·0	32·8
" G "	19,250	135,500	46·4	1·053	48·54	15·0	134·0	32·8
" H "	—		—	—	—		—	—
" I "	18,920	145,000	46·1	1·048	48·31	14·7	145·0	36·0
Heavy	18,030	15[illegible],600	46·66	1·04	48·52	13·8	136·0	35·7
Kerose	19,100	155,200	46·14	1·06	48·91	15·0	108·0	26·1
Pentan	19,740	123,100	46·25	1·051	48·7	15·25	154·0	37·8
Hexane	19,390	132,900	46·0	1·051	48·35	15·2	156·0	37·8
Heptan	19,420	134,100	46·06	1·056	48·61	15·1	133·0	32·4
Benzen	17,460	154,200	46·9	1·013	47·51	13·2	172·0	46·8
Toluene	17,660	153,600	46·9	1·023	47·98	13·4	151·0	40·5
Xylene	17,930	154,500	46·7	1·03	48·1	13·6	145·0	38·7
Cyclohe	18,940	149,000	46·08	1·044	48·11	14·7	156·0	38·7
Hexahy	18,890	147,200	46·01	1·047	48·2	14·7	138·0	34·2
Hexahy	18,890	140,600	46·1	1·054	48·59	14·8	133·0	32·4
Cracked	18,540	140,200	47·0	1·054	49·54	14·8 (App.)	150·0 (App.)	37·5
A								
Ethyl A	11,840	94,500	44·5	1·065	47·39	8·9	406·0	153·0
"	11,130	92,000	44·0	1·065	46·86	8·4	412·0	176·0
Methyl	10,030	83,300	45·5 (App.)	1·06 (App.)	48·2 (App.)	6·5	500·0 (App.)	252·0
Methyla	10,580	86,900	44·0	1·064	46·82	8·0 (App.)	450·0 (App.)	198·0
Butyl A	—	—	—	—	—		—	—
Ether (	16,830	122,500	46·4	1·06	49·2	13·0	146·0 (App.)	39·6
Carbon	10,730	106,600	40·2	0·98	39·4	10·8	146·0	48·6

primary importance, but every effort was made to ensure mechanical consistency.

(2) Every known expedient was adopted to attain the highest possible thermal efficiency and power output, and to ensure that all losses, whether thermal or frictional, were reduced to the absolute minimum, and maintained as nearly constant as possible, under all conditions.

(3) The engine was designed to run when required at a piston speed in excess of that of existing engines.

(4) Means are provided for varying the compression of the engine over any range from 3·7 : 1 up to 8 : 1 while running at full power, and without disturbing any temperature, frictional, mechanical, or other conditions.

(5) The combustion chamber is so designed that its general form and ratio of surface to volume undergo the minimum of alteration when the compression is varied, and to this end a very long stroke is employed.

(6) Special means are adopted to render the engine as little sensitive to changes in the temperature of the lubricant as possible. Ball bearings are used wherever possible, in order to reduce variation in friction with different oil temperatures, and the water jacketing round the barrel of the cylinder is stagnant, and therefore quickly attains a constant temperature, independent of the temperature of the supply. This ensures that the piston friction, which is dependent upon the temperature of the oil on the cylinder walls, reaches a minimum in the course of a few minutes, and thereafter remains constant. The importance of retaining as nearly as possible the same general form of combustion chamber under all conditions of compression cannot be overestimated. Very misleading results have been obtained when the compression ratio has been varied by fitting different pistons, some with concave, others with convex crowns. In one series of experiments with different compression ratios which the writer examined, and which were obviously carried out with the most scrupulous care, the results were entirely vitiated because the whole character and efficiency of the combustion chamber were completely changed, as between the low compression and the high, with the result that a certain optimum compression ratio was claimed to have been found, after which any further increase in compression resulted in loss of power and efficiency. A careful scrutiny of the results showed that at or near the so-called optimum compression ratio the efficiency of the combustion chamber

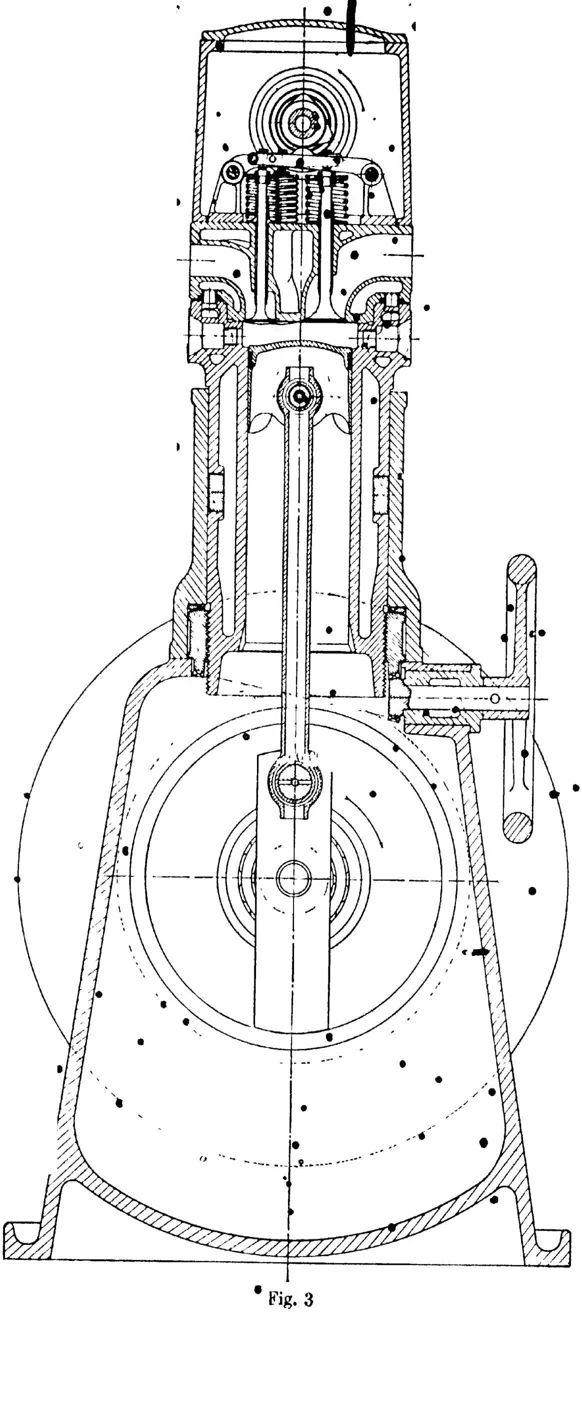


Fig. 3

	Mean Pressure at Useful Compression. 85 Min lb. per sq. in.	K. Max. Ind Mean Effective Pressure at Highest Useful Compression. No Heat lb per sq in	L. Rise or Fall of Temp in Induction Pipe (indicating app. Mean Volatility) Heat 65 B Th U /Min Deg F	M. Relative Thermal Efficiency (compared with that obtained with Toluene) At the same Compression Ratio per cent	M. At Highest Useful Compression per cent	N. Relative I M E P (compared with that obtained with Toluene) At the same Compression Ratio per cent	N. At Highest Useful Compression per cent
A " " " " " " " " "		138 1 148 5 146 0 142 4 142 9 136 7 140 5 135 4 147 8 132 5	+18 0 +19 8 +5 4 +12 6 +19 8 +18 9 +9 5 +32 4 +21 6 +25 2	100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) — 100 0 (App) —	83 7 93 0 91 0 86 7 88 4 81 8 85 6 — 92 2 —	100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App) 100 0 (App)	88 4 95 3 93 5 91 0 91 7 87 5 90 2 86 6 94 8 85 0
H K		— —	+50 5 +56 0	86 0 80 0	84 0 61 2	100 0 (App) 99 100	96 9 83 6
F I H	3·8 . 1	— 141 2 125 5	— 0 +10 0	— 100 0 (App) 100 0 (App)	— 86 4 71 2	100 0 (App) 100 0 (App) 100 0 (App)	94 5 90 5 80 5
F T	7 0 1 7 0 1	156 0 156 3 156 1	+13 1 +11 4 +32 4	(App) 100 0 (App) 100 0 (App)	99 2 100 0 99 5	100 0 (App) 100 0 (App) 100 0 (App)	99 7 100 0 99 9
C H H		148 0 146 6 138 1	− 5 4 + 5 4 +24 3	100 0 (App) 100 0 (App) 100 0 (App)	93 0 91 5 84 0	100 0 (App) 100 0 (App) 100 0 (App)	94 6 93 9 88 4
C		145 4	+10 0	100 0 (App)	84 0	100 0 (App)	92 5
E M M H E C	7 5 . 1 7 5 . 1	165 5 170 0 153 9 165 0 164 5 132 5 136 3	+ 2 7 + 3 6 −15 3 − 1 8 +18 0 + 1 8 −12 6	102 0 102 5 103 0 102 5 — — —	107 9 108 1 93 5 102 7 — — —	105 0 108 0 110 0 110 0 105 0 103 5 94 7	106 4 109 8 99 7 105 8 106 0 85·0 85·5

he efficiency and power obtained relatively to the other fuels if used at the same compression.

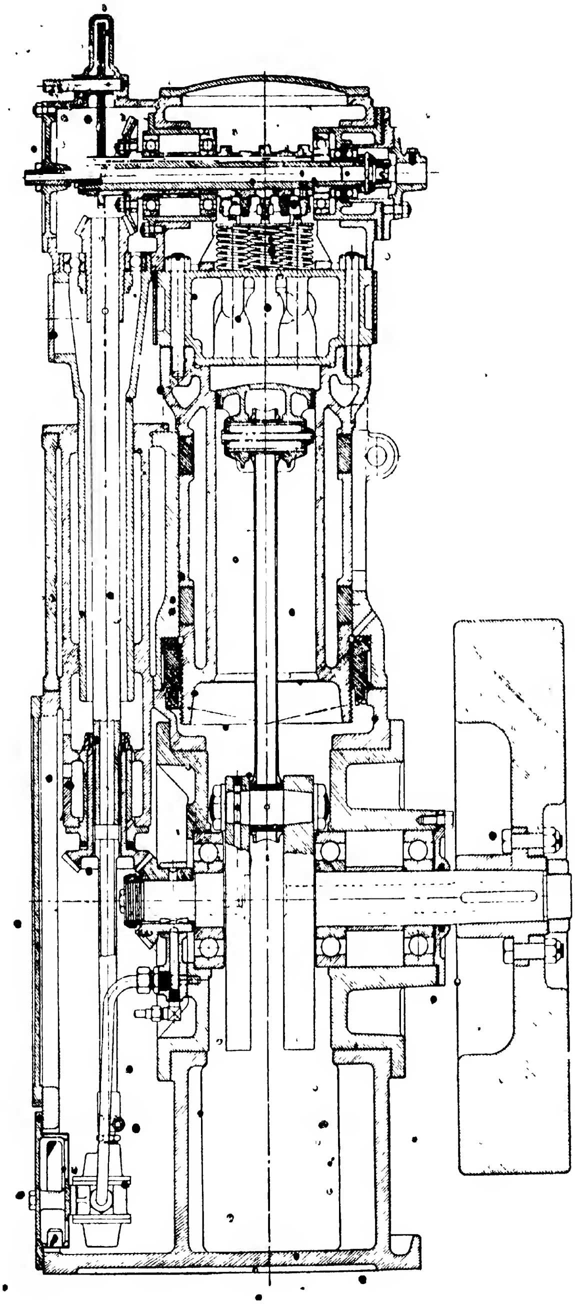


Fig. 4.—Variable Compression Engine, Sectional Side Elevation

was at a maximum, and that at the higher compressions it fell away very rapidly, and, indeed, became quite exceptionally inefficient.

In the variable compression engine designed for the purpose of these tests, the efficiency of the combustion chamber undergoes very little change between the lowest compression ratio and the highest, with the result that the efficiency increases with increase of compression at a perfectly steady rate throughout the whole range.

As will be seen from the sectional drawings, figs. 3 and 4, the compression ratio is varied by raising or lowering the whole cylinder, together with the carburettor, camshaft, and valve gear (fig. 5); by this means the compression ratio can be varied over any range in the course of a few seconds, and without disturbing any of the temperature conditions or any adjustments.

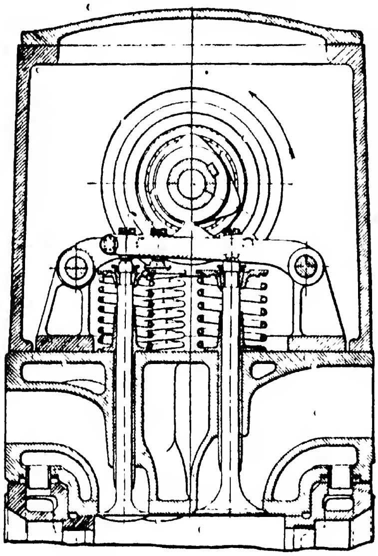


Fig. 5.—Section through Cylinder Head showing Valve Gear

To measure and record the compression ratio in use, a micrometer is provided as shown in fig. 6. This is arranged to operate electrical contacts, and controls a pilot lamp, which lights up immediately the desired compression is reached, so that the operator can adjust the micrometer screw at his leisure to the compression ratio he requires before making any alteration, and can then see at a glance by the lighting up of the lamp that this compression has been reached.

For the ignition of the charge four sparking plugs are fitted equidistant round the circumference of the combustion chamber, each of which is connected to a Remy high-tension coil. The low-tension circuit of all the coils is operated by a single Remy contact breaker driven directly from one end of the camshaft. The object of using this arrangement in preference to magnetos was twofold :

(1) To ensure that the passage of the spark across all four plugs should be absolutely synchronous.

(2) To ensure that the intensity of the spark should be the same at all settings.

In practice, it was found that the use of two sparking plugs on opposite sides of the combustion space gave equally good results, and all tests were therefore run under these conditions. To measure accurately both the power and friction losses the engine is direct coupled to a balanced swinging field electric dynamometer, one arm of which carries a dead weight of 40 lb., which is slightly in excess

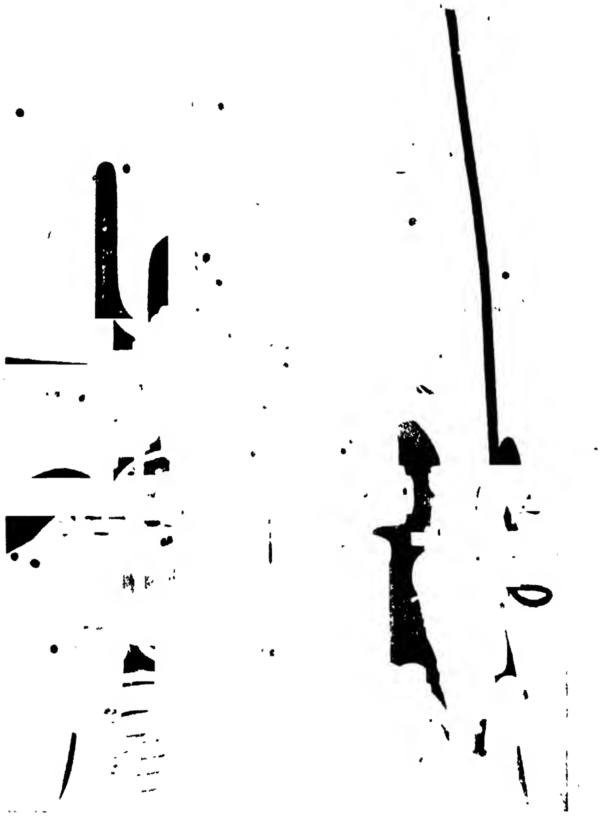


Fig. 6.—Photo of upper Part of Engine showing Micrometer for measuring Compression Ratio

of the maximum torque of the engine—a light open-scale spring balance is used to record the difference in torque between the dead weight and that developed by the engine—this arrangement permits of exceedingly accurate determinations, since a very small variation in torque corresponds with a wide range on the spring balance. The mean torque on the dynamometer arm is in the neighbourhood of 35 lb., and the difference can be read off at a glance to within less

than one-tenth of a pound. The steadiness of the dynamometer is such that the needle of the spring balance does not vibrate or oscillate through a range of more than ±0·1 lb. Generally speaking, all readings of torque may be taken as being accurate to within one-third of 1 per cent, while the standard of accuracy of the average of

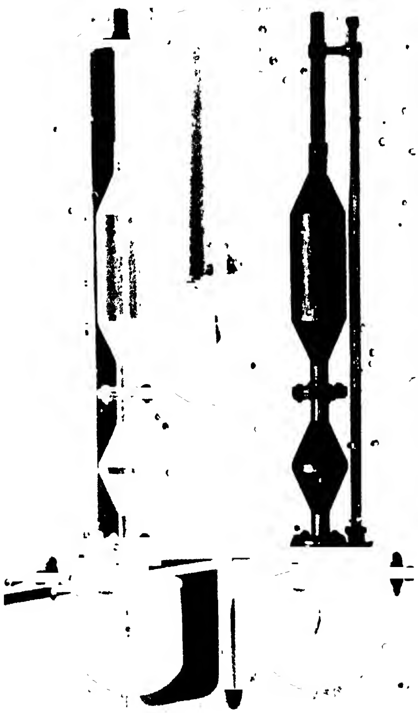


Fig. 7.—Fuel-measuring Device

several readings is, of course, considerably higher. The load is controlled by varying the field excitation of the dynamometer. For this purpose, two rheostats are provided in the field circuit, one of which gives coarse graduations, and the other, a continuous coil resistance, affords continuous range, and is used for fine adjustments.

The fuel-measuring device consists of two vessels, each comprising

two conical-ended chambers connected together at either end by a narrow throat, as shown in fig. 7. The upper chamber has a capacity of exactly one pint, and the lower of ¼ pint. A gauge glass is fitted to each vessel. The rate of fall of liquid in the glass is very rapid when passing the narrow throats, so that its passage past the marks on the gauge glass can be clocked with extreme accuracy. Geared off one end of the camshaft is a revolution counter operated by means of a magnetic clutch, and so arranged that the counter is thrown into operation as the liquid in the gauge glass passes the first mark, and is thrown out, and a brake applied to its spindle as the liquid passes the second mark; thus the actual number of revolutions during the consumption of either one pint or ¼ pint of fuel is automatically recorded.

The carburettor is a standard Claudel-Hobson aircraft type, but fitted with a fine adjustment needle valve controlling the jet, so that the mixture can be varied between close limits. An electrical heater is fitted in the carburettor air intake passage, and the exact amount of heat supplied can be read off from instruments on the switchboard. A thermometer fitted in an insulated pocket, and projecting into the inlet valve port, is provided to record, very approximately, the temperature of the working fluid during its entry to the cylinder. From the known amount of heat supplied, and from the measured difference in the temperature of the air before and after its entry to the carburettor, it is possible to determine at least a relative measure of the mean volatility of the fuel used. The readings of the thermometer in the induction passage are, however, of relative value only. Owing to the variations in the temperature of the thermometer pocket due to the deposition of liquid fuel upon it, this thermometer behaves as a wet bulb instrument, and even the relative values of its readings cannot be relied upon as between fuels of widely different latent heats of evaporation.

A general lay-out of the apparatus is shown in fig. 8.

A small calibrated gas-holder is also provided, from which the engine can draw its supply of air when running on a liquid fuel.

The fall of the gas-holder controls an electrical contact mechanism which in turn operates a magnetic counter on the observer's desk; this counter being inter-connected, electrically, with the revolution counter on the engine. By this means the air consumption per revolution can be measured with great accuracy.

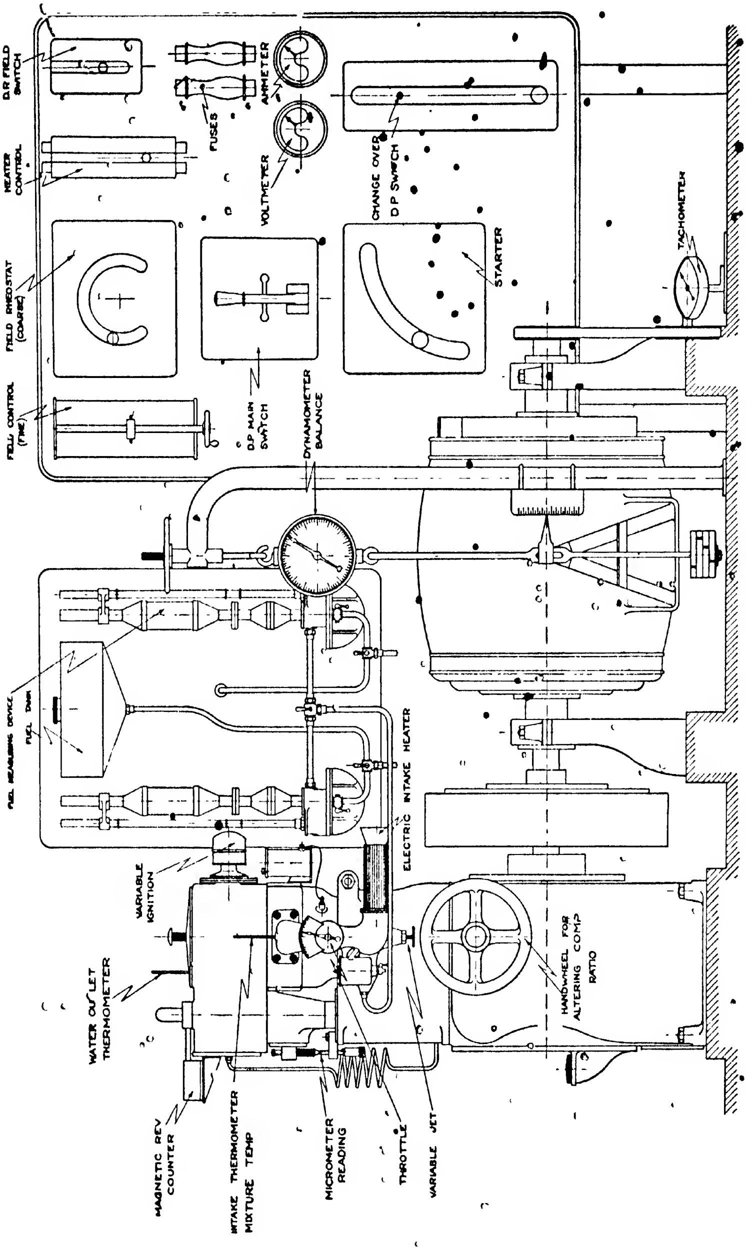


Fig. 8.—General Lay-out of Testing Apparatus

CHAPTER II

DETONATION

The phenomenon of detonation appears to be the setting up in the cylinder of an explosion wave. This occurs when the rapidity of combustion of that portion of the working fluid first ignited is such that, by its expansion, it compresses before it the unburnt portion beyond a certain rate. When the rate of temperature rise due to compression by the burning portion of the charge exceeds that at which it can get rid of its heat by conduction, convection, etc., by a certain margin, the remaining portion ignites spontaneously and nearly simultaneously throughout its whole bulk, thus setting up an explosion wave which strikes the walls of the cylinder with a hammer-like blow and, reacting in its turn, compresses afresh the portion first ignited. This further raises the temperature of that portion, and with it the temperature of any isolated or partially insulated objects in its vicinity, thus soon giving rise to pre-ignition. It would appear, therefore, pretty certain that detonation depends primarily upon the rate of burning of that portion of the charge first ignited, and it remains to discover what actually controls this rate.

It would seem that turbulence, while invaluable for other reasons, influences detonation but little one way or the other. In the case of combustion chambers designed to give very high turbulence, there is generally found a marked reduction in the tendency to detonate, but in such case this may be generally ascribed, not to the turbulence, but rather to the fact that in each instance the maximum distance which the flame could travel from the sparking plug was exceptionally small. More recent tests which the author has carried out on engines with multiple valves in which turbulence could be varied by cutting out one or more inlet valves showed that this made no difference whatever as regards detonation.

Until recently it was always supposed that detonation was dependent upon the temperature of compression; this appeared plausible enough at first sight, but it most certainly did not fit in

with the observed facts. For example, the difference in compression temperature between a compression ratio of 4 : 1 and 6 : 1 is actually about 70° F. only, and this difference can easily be more than counterbalanced by a change in the amount of pre-heating of the charge, but practical experience teaches that while detonation on, say, a reasonably good petrol will not occur even with excessive pre-heating at a compression ratio of 4 : 1, it will inevitably occur far below 6 : 1 without any pre-heating at all, and even with stone-cold cylinder jackets. In this connection the curve, fig. 9, shows the approximate compression temperature for a range of compression

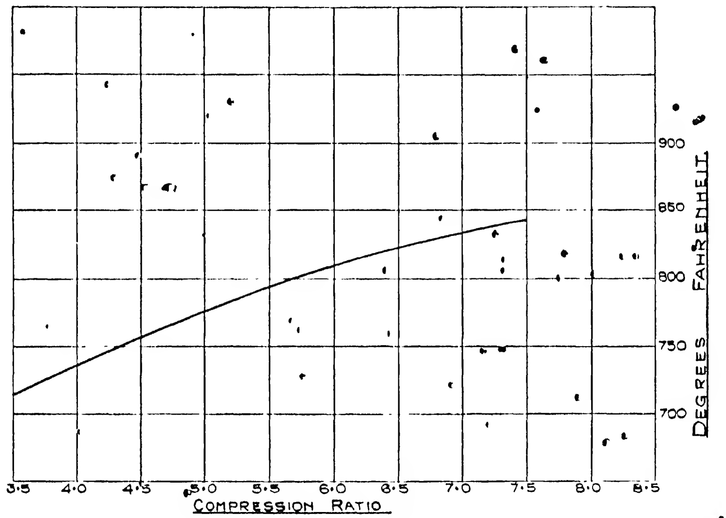


Fig. 9.—Compression Temperature for "Correct" Petrol-air Mixture

from 4 : 1 to 8 : 1, assuming (1) the same amount of pre-heating and the same latent heat in every case; (2) allowing for the varying proportion and temperature of the residual products at each compression ratio.

Careful experiments appeared to show pretty clearly that detonation had very little connection with the temperature of compression, but was closely dependent upon the compression pressure; and it was therefore supposed that it was the pressure, rather than the temperature of the working fluid, which controls the initial rate of burning, and therefore the tendency to detonate. This seemed reasonable; it fitted in nicely with the facts of the case, and did duty for a while as an explanation, until the chemists objected, on

the grounds that the rate at which combustion takes place under these conditions is generally influenced but little by relatively small differences of pressure. Quite recently Tizard and Pye have carried out a careful investigation into the phenomena of detonation, and while it is perhaps beyond the scope of this book to give their reasoning in full, their conclusions may be summarised as follows :

(1) Detonation depends in the first place, though not wholly, upon the rate of burning of that portion of the charge first ignited—in this they confirm the usually accepted theory.

(2) That the rate of burning increases very rapidly with slight increase of temperature, and that whether it will prove sufficiently rapid to produce detonation or not depends upon the ratio between the rate of evolution of heat by the burning portion of the mixture, and the rate of heat loss.

(3) The chance that the rate of burning of any portion of the mixture will become so high as to cause detonation depends, in so far as practical engine conditions are concerned, neither upon the temperature nor the pressure of compression, but rather upon the maximum flame temperature.

(4) For any given mixture strength, the maximum flame temperature depends primarily upon the proportion of diluent or exhaust products present—it depends also, of course, upon the compression temperature, but this varies very little over a wide range of compression ratio, while the variation in the proportion of residual exhaust products over the same range, exerts a much greater influence in diluting the charge, and so lowering the temperature of the flame. Thus, a difference of 1 per cent by weight of exhaust diluent will raise or lower the flame temperature by roughly about 40° F., equivalent to a range of compression ratio from 4 to 5 : 1.

(5) If the flame temperature be reduced by weakening the mixture strength, a very much higher compression could at once be used. In practice, with the exception of hydrogen, it is not possible so to weaken the mixture as to effect any appreciable reduction in flame temperature, because within the narrow range available, weakening the mixture with air results merely in reducing the amount of dissociation without affecting appreciably the flame temperature. Fig. 10 shows the observed variation in compression ratio permissible over a wide range of mixture strength. In this experiment the engine was run with wide-open throttle at constant speed and constant temperature, and the compression ratio adjusted at each mixture strength until detonation just became apparent.

Both with hydrogen as a weak homogeneous charge, and with petrol in a stratified charge, it was found possible to operate with a compression ratio of about 7·0 : 1 with a mean mixture strength 50 per cent weak in either case, and that without the least trace of detonation. Fig. 11 shows a similar test with hydrogen, but with the mixture strength much further reduced, as is possible only with this fuel. On the other hand, with hydrogen, the range of burning on the rich side could not be explored because, so soon as any excess

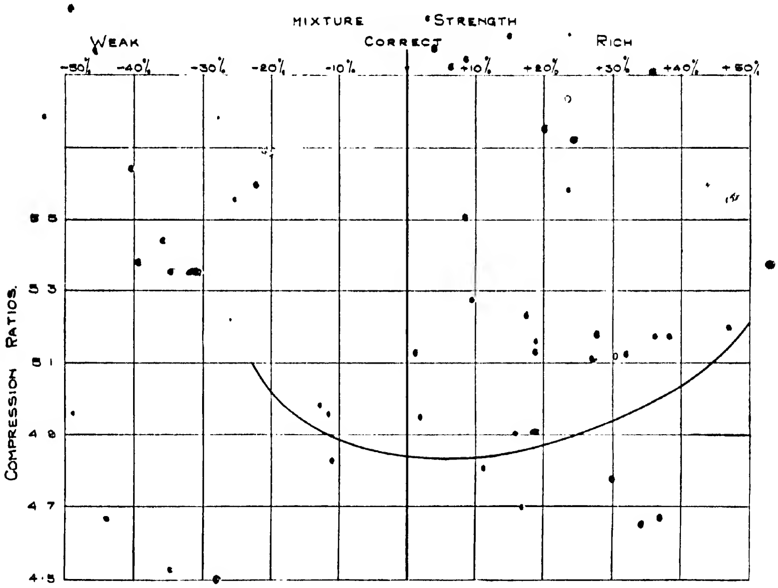


Fig. 10.—Compression Ratio at which Detonation becomes apparent with varying Mixture Strength using Aromatic Free Petrol, and with all other Conditions constant

of hydrogen was admitted, back-firing occurred through the inlet valves.

It will be seen that, in the case of a normal homogeneous mixture, the tendency to detonate depends, in effect, upon the compression pressure, not, as was supposed, because the pressure exerts a controlling influence, but rather because, in any actual engine, the compression pressure is, in itself, a measure of the proportion of inert exhaust gas left behind in the cylinder. It differs in practice only when, either by the use of hydrogen, by stratifying the charge, or by the introduction of inert diluents, a weak mean mixture strength can be used. Tizard's theory has been further confirmed by other tests:

(*a*) In which the residual exhaust products have been cleared

away by scavenging with air, when it was found that detonation at once became severe, even with very low compression pressures.

(*b*) In which additional exhaust products were added by way of the carburettor, when the compression could be raised to almost any degree depending upon the quantity admitted. Fig. 12 shows the variation in compression ratio permissible when, to a petrol detonating normally at a compression ratio of 4·85 : 1, varying quantities of additional exhaust products were added and the compression adjusted in each case until detonation just became apparent. The

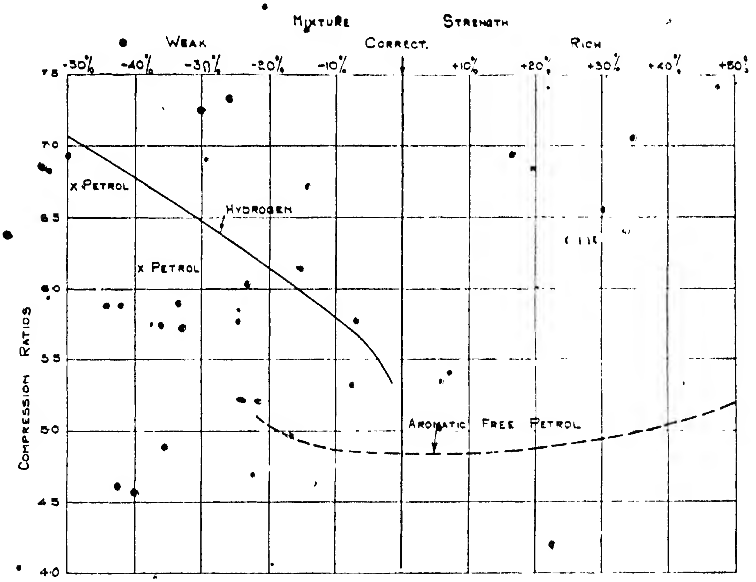


Fig. 11.—Compression Ratio at which Detonation becomes apparent with varying Mixture Strength. Full line, Hydrogen. Dotted line, Aromatic Free Petrol, also two Points obtained with Aromatic Free Petrol when working with a stratified Charge (X)

effectiveness of such inert gases appears to be closely proportional to their specific heats, that is, to their direct influence upon the flame temperature.

Influence of the Nature of the Fuel upon Detonation.—Broadly speaking, it would appear both from actual engine tests and from Tizard's researches, that two factors determine whether or not a fuel will detonate :

(1) The self-ignition temperature of the fuel/air mixture.

(2) The rate of acceleration of burning as the ignition temperature is exceeded.

Both the true self-ignition temperature, if indeed such a term can be used, and the rate of acceleration of burning appear to depend primarily upon the chemical composition of the fuel.

Reference has been made to the self-ignition temperature of a fuel/air mixture. Strictly speaking there is no such thing, for any fuel/air mixture will actually combine in time at almost any temperature. In the case of hydrocarbon fuels the rate of combination

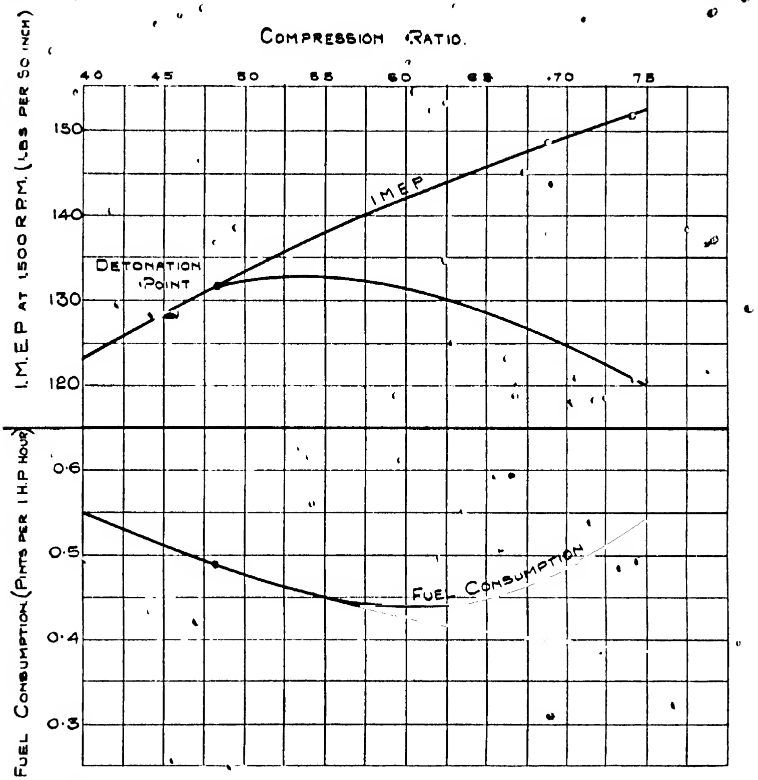


Fig. 12.—Curves showing Fuel Consumption and I.M.E.P. with cooled Exhaust Gas added.

is multiplied about three times for every 3 per cent increase in temperature, so that, in practice, there is a comparatively narrow range of temperature over which the rate of burning is reasonable. In this connection the case of carbon disulphide is particularly interesting—this fuel has an exceedingly low ignition temperature, so low that when used neat in an engine it will pre-ignite at once even at a compression ratio of 3·6 : 1, yet, when mixed with petrol, carbon disulphide is quite effective in preventing detonation and will allow of the use

of a higher compression ratio than is normally possible with petrol. On investigation it has been found that, in the particular case of carbon disulphide, the rate of acceleration of burning is very much slower and that it trebles its rate for every 7 per cent rise in temperature as against about every 3 per cent for all hydrocarbon fuels.

Range of Burning.—So long as a fuel contains no free hydrogen, the available range of burning differs but little between the different volatile liquid fuels and is, compared with illuminating or other gaseous fuels, very narrow indeed. It is clear that so far as efficiency is concerned, it is only the range of burning on the weak side that need be taken into consideration; the range of burning on the rich side, that is up to the point at which the mixture fails to ignite owing to over-richness, is of comparatively little practical interest. It has already been shown that any range on the weak side is of the utmost importance, for so long as the fuel will burn completely, the weaker the mixture, the lower the flame temperature, and therefore the higher the efficiency. If combustion were complete the flame temperature would diminish very nearly in proportion to the mixture strength. Unfortunately, incomplete and delayed combustion become apparent so soon as the mixture strength is reduced by more than about 15 per cent below that giving complete combustion; from 12 per cent to 18 per cent weak, the loss by delayed and incomplete combustion and the gain due to the lower flame temperature, just about balance each other, while beyond about 18 per cent weak the net efficiency begins to fall away rapidly, and the rate of burning becomes so slow that it continues throughout the exhaust stroke and so ignites the fresh charge on its entry to the cylinder, causing the familiar back-fire into the induction system. This can, however, be obviated to a limited extent by advancing the time of ignition, and the range on the weak side can be extended slightly in consequence, but the angle of ignition advance required, efficiently to burn a mixture only 20 per cent weak, is so excessive as, in practice, to be almost impracticable. In fig. 13 is shown a typical curve showing the relation between thermal efficiency, mean pressure and mixture strength with fixed ignition timing. The mixture strength is plotted in terms of percentage excess of fuel or air. In fig. 14 is shown a similar curve, but with the ignition advanced a further 30° between 10 per cent and 20 per cent weak; in both cases the mean pressure and efficiency were obtained by direct measurement, and represent the mean of a large number of tests on various fuels. The point at

which combustion is complete, that is, at which there is no excess either of fuel or air, is shown by a vertical line, to the right of which

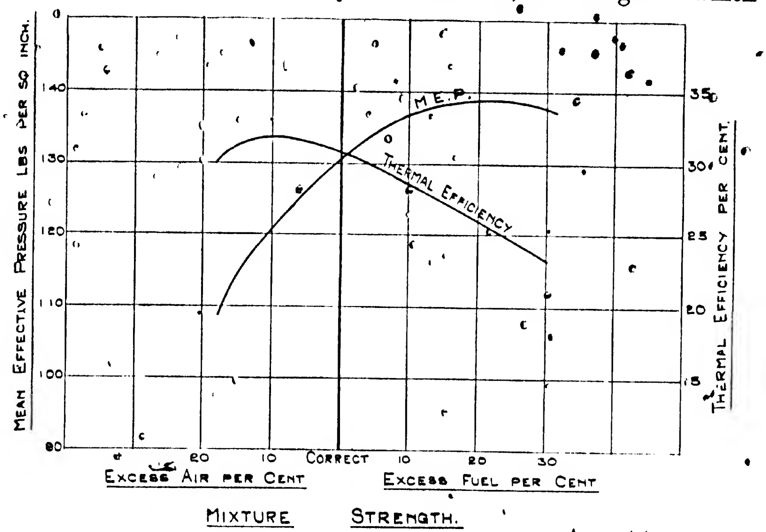


Fig. 13.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with fixed Ignition Timing. Fuel, Petrol

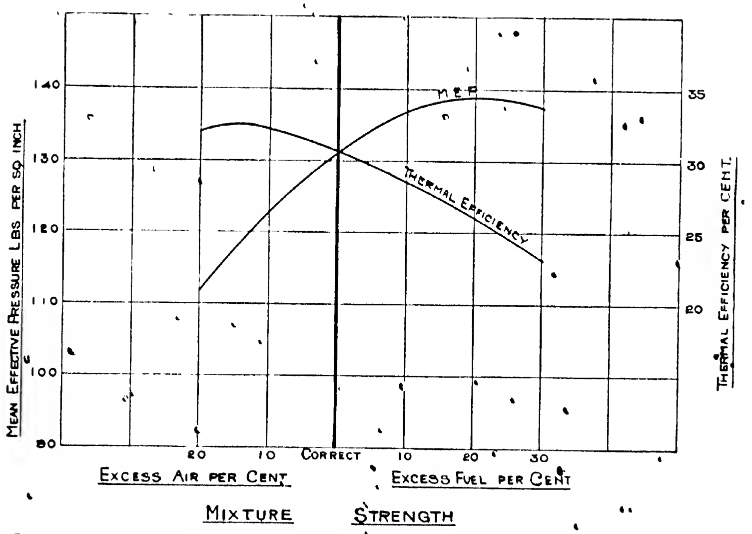


Fig. 14.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Petrol

the mixture is over rich and to the left it is weak. In fig. 15 is shown a similar curve taken with alcohol. It will be observed that while,

in each case, the M.E.P. increases as the mixture is enriched beyond the point of complete combustion, it is only in the case of alcohol that there is any increase beyond 20 per cent excess of fuel. The increase in mean pressure with excess fuel depends upon the inter-relation of a number of factors; on the one hand are the increase in specific volume and the increase in volumetric efficiency due to the latent heat of the fuel, both of which tend to increase the power output with further increase of mixture strength. Against these must be offset the higher specific heat of the products of combustion, when the fuel is only partially burnt. In the

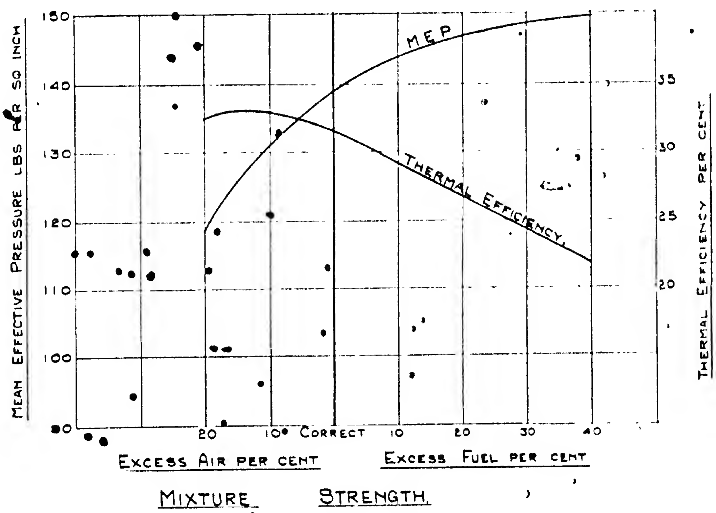


Fig. 15.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Alcohol

case of petrol and benzol the increase is small, because the latent heat is low and the several factors more or less balance one another. The latent heat of benzol is higher than petrol, and moreover more use can be made of it, because the fuel is homogeneous, but, on the other hand, the change in specific volume is smaller. In the case of alcohol, the latent heat is very much higher and the change in specific volume also is greater, consequently with this fuel the mean pressure increases as the mixture is enriched to a far greater extent than with either of the other two.

The range of burning on the weak side is substantially the same in all three cases and indeed in the case of all known volatile liquid fuels.

It will be observed that the efficiency is at a maximum when the excess of air is between 10 and 18 per cent; with an excess of 20 per cent the process of combustion becomes so slow that the running of the engine is unstable, and both misfiring and back-firing through the inlet valve are liable to occur. The tests on which these results are based were all carried out on single-cylinder engines. In the case of a multi-cylinder engine, however well designed the distribution system may be, it is practically impossible to maintain the mixture strength to all cylinders uniform to within closer limits than about 10 to 20 per cent. If the mean strength supplied by the carburettor is, say, 15 per cent weak, it follows that one or more cylinders will receive mixture as much as 20 to 25 per cent weak which will result in misfiring and back-firing, through the carburettor. To allow therefore for inequalities in distribution, it is necessary, in practice, even in the very best examples to supply a mixture containing not more than about 10 per cent excess of air. With such a mixture the supply to some cylinders will be about that giving complete combustion to others about 20 per cent weak, and the maximum efficiency obtainable will be about 1·5 per cent lower than could be obtained from a single-cylinder engine. It will be seen, therefore, that, in order to legislate for the weakest cylinder and prevent back-firing, one or more of the remainder must receive a mixture which is too rich for maximum economy, and the indicated thermal efficiency of a multi-cylinder engine will, on this account, always be lower than that of a single cylinder. Further, the greater the number of cylinders fed from any one source of supply, the lower the efficiency.

To sum up, (1) the available range of mixture strength on the weak side, with all volatile hydrocarbon fuels, is very narrow, far narrower than with most of the gaseous fuels.

(2) Owing to the narrow range of available mixture strength and to the inevitable inequalities in distribution, multi-cylinder engines cannot run with the most economical mixture strength, hence their fuel consumption per H.P. hour must always be slightly higher than that of a single cylinder—how much higher, depends upon the number of cylinders supplied by any one carburettor and, of course, on the efficiency of the distribution system.

(3) With all fuels, slightly more power is developed with an over-rich mixture as compared with the mixture giving complete combustion; with alcohol the increase in power is very marked, and amounts to nearly 10 per cent with very rich mixtures.

(4) In the case of single-cylinder engines running on petrol or benzol, maximum economy is obtained at about 92-94 per cent of full power; in the case of multi-cylinder engines, for the reasons given above, the maximum economy on the same fuels is slightly lower, but is obtained at about 96 to 97 per cent of full power.

The Temperatures of the Cycle.—From the data and information now available, it is possible to deduce with a fair degree of accuracy, the temperature changes throughout the cycle when liquid hydrocarbon fuels are used.

The first temperature which has to be determined is that at the end of the suction stroke, for from this all the other cycle temperatures can be obtained. It is also of direct importance as determining the volumetric efficiency of the engine.

The temperature at this point is influenced by the final exhaust temperature, owing to the admixture of the fresh charge with the residual exhaust products in the clearance space. It therefore should strictly be evaluated by a "hit-and-miss" type of calculation, by which a residual exhaust temperature is assumed, and the cycle temperatures worked out on that assumption. The assumed exhaust temperature is then modified till it agrees with that which is obtained by calculating round the cycle.

Fortunately, however, it requires a large change in exhaust temperature appreciably to alter the suction temperature, so that the assumed and calculated temperatures need not check very closely.

It is, in the author's opinion, always preferable to take a concrete example; we will therefore consider the case of a cylinder with a swept volume of 80 cub. in. running at 2000 R.P.M. Let the compression ratio be 5 : 1, making the total cylinder volume 100 cub. in., and let the following conditions be assumed:—

Mean jacket temperature	140° F.
Heat input to charge external to cylinder (by carburettor heating, &c.)	0·05 B.Th.U. per cycle.
Absolute pressure in cylinder at end of suction stroke	14·0 lb./in.
Absolute pressure in cylinder at end of exhaust stroke	14·7 ,,
Mean temperature of outside air	60° F.

These are conditions which may be taken to represent average practice.

The fuel is assumed to be an average high-class petrol of composition say 50 per cent paraffins, 35 per cent naphthenes, and 15

per cent aromatics, while the case of ethyl alcohol and benzene (benzol) are also dealt with.

The relevant properties of these fuels are :

Fuel.	S.G.	Effective Cal Val. for B.Th.U./lb.	Latent Heat of Evaporation, B.Th.U./lb.	Boiling Point.
Petrol	0·740	19,000	135	160°-400° F.
Benzene	0·884	17,460	172	176° F.
Ethyl alcohol ...	0·798	11,840	397	172° F.

Fuel.	Change in Sp. Vol after Combustion (Correct Mixtures)	Complete Combustion Mixture, (by Weight).	Energy liberated per Standard cub. in of Mixture ft. lb. exclusive of change of Specific Volume.
	per cent.	air	
Petrol	+5·0	14·3/1	46·2
Benzene	+1·3	13·2/1	46·9
Alcohol...	+6·5	8·95/1	44·5

Consider first the conditions when running on petrol, starting with the commencement of the suction stroke. The cylinder then contains 20 cub. in. of hot exhaust products at atmospheric pressure. These will be at a temperature of about 2100° F. absolute, as will be shown later. It is desirable in order to arrive at their heat capacity relative to that of the entering charge to reduce them to terms of normal temperature and pressure. The volume of the residual exhaust products under these conditions will therefore be

$$20 \times \frac{491}{2100} = 4{\cdot}68 \text{ cub. in.}$$

The incoming charge consists of air at a temperature of about 60° F. and a small proportion of fuel, entirely or partly vaporized The air/fuel ratio for complete combustion will, in the case of petrol be 14·3/1. The latent heat of evaporation of 1 lb. of petrol is 135 B.Th.U.s. This is supplied by 14·3 lb. of air of which the specific heat at constant pressure is 0·237 ; the drop in temperature of this air will be $\dfrac{135}{14{\cdot}3 \times 0{\cdot}237} = 40°$ F.

For the mass of air in question 1° F. change in temperature is produced by ·00067 B.Th.U., so that the latent heat of evaporation

will take up 40 × ·00067 = ·00268 B.Th.U. From the cylinder walls at a temperature of about 140° F., and from the still hotter valve and piston surfaces, &c., the charge picks up about 0·0005 B.Th.U. per cub. in., or in this case 0·04 B.Th.U. per cycle. The net gain of sensible heat to the charge is 0·04 + 0·05 − 0·0268 = 0·0632 B.Th.U. per cycle, which will raise the temperature by about 95° F. to 155° F., assuming that all the fuel is completely evaporated. If the direct heating does not complete evaporation, mixture with the hot residual gases will certainly do so unless the mechanical pulverization of the fuel in the carburettor is very inadequate. It follows therefore that

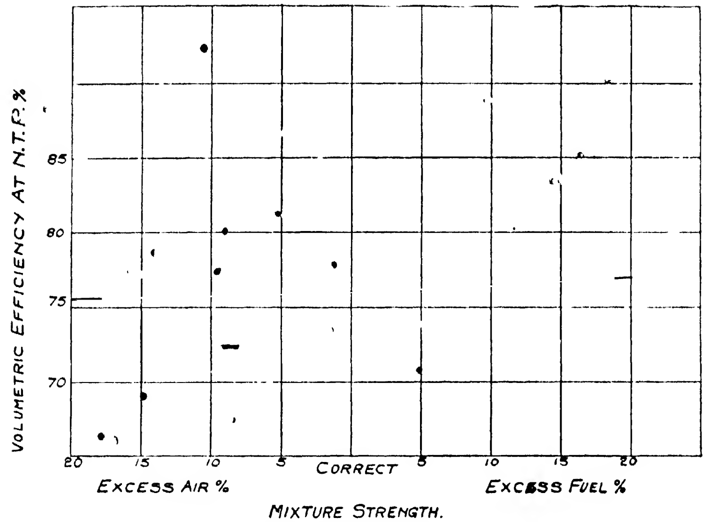


Fig. 16.—Observed Volumetric Efficiency with different Mixture Strengths at a Compression Ratio of 5·0 : 1. Fuel, Petrol

the final suction temperature will be the same at whatever stage the latent heat of evaporation is extracted, provided only that evaporation is complete before compression starts.

We now have 80 cub. in. of fresh charge at 155° F. or 614° F. abs. and 14·0 lb./in. absolute pressure. Reduced to N.T.P. this becomes $80 \times \frac{14\cdot0}{14\cdot7} \times \frac{491}{614} = 60\cdot9$ cub. in. This gives a volumetric efficiency of $\frac{60\cdot9}{80} = 76\cdot2$ per cent, a figure which agrees very closely with that found in, for example, the author's single-cylinder research engine under similar temperature conditions, as shown in fig. 16, which indicates the observed volumetric efficiency at different mixture

strengths when running with the heat input to the carburettor specified above, and at a piston speed of 2000 ft. per minute.

The volume of the residual exhaust gases at N.T.P. was shown to be 4·68 cub. in., so that the total standard volume of the combined mixture $= 60 \cdot 9 + 4 \cdot 68 = 65 \cdot 58$ cub. in. As this actually fills a volume of 100 cub. in. at a pressure of 14·0 lb./in. its temperature must be $\frac{100 \times 14 \cdot 0}{14 \cdot 7 \times 65 \cdot 58} \times 491 = 717^\circ$ F. absolute or 258° F.

This figure may be taken as the final suction temperature to within about $\pm 10^\circ$ F. in the case considered. The chief possibilities of error lie in

(1) The temperature of the exhaust products. Owing, however, to the fact that as their temperature increases their mass correspondingly decreases, a large error here has but little influence on the suction temperature.

(2) The heat picked up from the cylinder walls, &c. This is certainly rather a doubtful figure, but the author has arrived at it by running an engine with varying jacket temperatures, and adding known quantities of heat by means of an electrical resistance in the induction pipe until the volumetric efficiency, and therefore the suction temperature, was constant. By measuring the difference in external heat necessary to do this with varying jacket temperatures, it was possible to estimate the total amount of heat taken up from this source.

(3) In most cases the amount of pre-heating of the charge before its entry to the cylinder is very uncertain, since in practice the heat is generally supplied either from the exhaust or from the circulating water. For experimental work the author prefers to supply this heat electrically so that it can readily and accurately be measured.

If benzene be used instead of petrol, we find that, owing both to its higher latent heat and to the greater proportion of fuel needed to combine with the air, the final suction temperature is lower, namely, about 235° F., while the volumetric efficiency is correspondingly higher, about 78·5 per cent. The residual exhaust temperature may be taken, as with petrol, as 2100° F. absolute.

In the third case, that of ethyl alcohol, the extremely high latent heat and the large proportion of fuel in the mixture produce much greater cooling effect, with the result that, arguing on the above premises, the final suction temperature, even after admixture with the residual exhaust products, will only be 67° F.; the corresponding volumetric efficiency should therefore be 104·3 per cent.

At such a temperature, however, and in the short time available, the alcohol would not be evaporated completely by the end of the suction stroke, so that such a calculation is not applicable in this instance. Experiment has shown that the minimum temperature at which evaporation is completed before compression, is in the neighbourhood of 150° F., and it will be well to consider the case in which sufficient pre-heating is supplied to effect this. In this case the volumetric efficiency works out at 90 per cent, and 37 per cent of the fuel remains to be evaporated by means of additional external heating to the extent of $\cdot 37 \times 0\cdot 143 = \cdot 053$ B.Th.U. per cycle.

In practice it is found that the volumetric efficiency on alcohol is considerably lower, namely, 82-83 per cent as compared with 76 per cent in the case of petrol. This shows that under normal working conditions only a comparatively small proportion of the alcohol is actually evaporated by the end of the suction stroke. It is mainly for this reason that efficient pulverization, which accelerates the transfer of heat from the air to the fuel particles, is so particularly desirable when using alcohol.

The above conclusions may be summarized as follows. Under the conditions quoted above—and they may be taken as average working conditions, except perhaps in the case of alcohol the final suction temperature and the volumetric efficiency will be :

Fuel.	Suction Temperature.	Volumetric Efficiency.
	° F.	per cent.
Petrol	258	76·2
Benzene...	235	78·5
Alcohol	150	90·0

With a 20 per cent weak mixture the figures become :

Fuel.	Suction Temperature.	Volumetric Efficiency.
Petrol	263	75·5
Benzene...	242	77·6
Alcohol	150	89·6

In the case of alcohol extra heat to the amount of 0·033 B.Th.U.

per cycle has to be supplied, and 0·029 B.Th.U. with the weak mixture, if evaporation is to be complete.

It should be noted that the volumetric efficiency refers to the volume of air and fuel vapour, not air alone.

Compression Temperature.—During the compression stroke the fuel and air are compressed nearly adiabatically to one-fifth of their original volume. At the commencement of compression the mixture, in the case of petrol or benzene, is slightly below the mean surface temperature of the cylinder, so at first they will absorb heat, but will lose heat later in the stroke. For these fuels the mean value of the exponent during compression for such an engine running at 2000 R.P.M. may be taken as 1·35. With alcohol the conditions are not quite the same, for

(1) The final suction temperature is substantially below the mean wall temperature, so that the mixture is appreciably heated during the earlier part of the stroke.

(2) As the alcohol forms an appreciable fraction of the charge (11 per cent by weight), it will decrease the mean exponent γ for the mixture, since γ for alcohol vapour is only 1·13 as against 1·4 for air.

(3) A considerable portion of the fuel is liquid at the end of the suction stroke, unless extra induction heat is supplied, and some of the heat of compression is applied to its evaporation.

We will ignore for the moment the last factor, and take γ for a "correct" alcohol mixture as being about 1·33.

The absolute temperature at the conclusion of the compression stroke in the case of petrol or benzene may be arrived at by multiplying the final absolute suction temperature by $5^{(1 \cdot 35 - 1)} = 1 \cdot 755$; This gives:

Petrol	correct mixture	1258° F. abs.
	20 per cent weak mixture	1267° F. abs.
Benzol	correct mixture	1220° F. abs.
	20 per cent weak mixture	1232° F. abs.

The compression pressure in each case will be $14 \cdot 0 \times 5^{(1 \cdot 35)} = 123$ lb. per square inch absolute or 108·3 lb. per square inch gauge.

With the alcohol mixtures, the final suction temperatures are the same both for the normal and the weak mixtures, so that the compression temperature, uncorrected for the latent heat of the surplus liquid, are also the same, being

$$609 \times 5^{(1 \cdot 33 - 1)} = 1035° \text{ F. abs.}$$

With the correct mixture there is, as previously shown, a heat deficit of 0·053 B.Th.U. per cycle, this amount being necessary for the evaporation of the alcohol which remains in a liquid state at the end of the suction stroke. This causes a drop of $0{\cdot}053 \times 1270 = 67°$ F., making the actual compression temperature 968° F. abs. or 475° F.

With the weak mixture, the heat deficit is only 0·029 B.Th U. per cycle, which raises the compression temperature to 998° F. abs. or 505° F.

The uncorrected compression pressure is $14{\cdot}0 \times 5^{(1{\cdot}33)} = 119$ lb. per square inch absolute. The cooling due to evaporation reduces this to 111 lb. per square inch abs. for the normal mixture, and 114·7 lb. per square inch abs. for the weak mixture.

Combustion Temperatures.—Since combustion takes place at constant volume, the whole of the chemical energy stored in the mixture, with the exception of that lost to the walls of the combustion chamber, will be devoted to increasing its internal energy. It is well known, however, that the obvious method of finding the temperature rise, that of dividing the heat output by the specific heat at normal temperatures, gives results that are far too high, in fact practically double those that are actually attained. This discrepancy is due to the three following influences:—

(1) The specific heat of the gases, of which the working fluid is composed, increases with increase of temperature.

(2) At high temperatures the products of combustion, CO_2 and water, dissociate with absorption of heat into carbon monoxide and oxygen, and hydrogen and oxygen respectively, according to the formulæ

$$2CO_2 \rightleftharpoons 2CO + O_2$$
$$2H_2O \rightleftharpoons 2H_2 + O_2.$$

(3) Heat is lost by radiation and conduction to the walls of the combustion chamber.

The specific heat of the gases, of which the working fluid is composed, and the amount of dissociation at any temperature, have been found by direct experiment, while the loss to the jackets can only be estimated from actual engine experiments. The amount of energy taken up by the gases as sensible heat can readily be calculated, since the specific heat of a gas at constant volume is unaffected by changes of pressure or by admixture with other gases. Unfortunately the degree of dissociation is influenced by alterations in pressure and, in cases such as the present, where two kinds of

dissociation have a common product (oxygen in this case), by the relative proportions of the gases present. This renders the actual calculation of the energy absorbed in this manner a somewhat complex and extremely wearisome process; it has however, been dealt with very fully by Tizard and Pye, so that the results alone will be given. The most convenient method is to construct a curve showing the total internal energy of the mixture, at any temperature, plotted against the temperature. Since the amount of internal energy at any stage in the cycle can easily be found, the corresponding temperatures can at once be read off.

From the work of Tizard and Pye[1] which is based on the experimental data of Pier and Bjerrum[2] the curve shown in fig. 17 has been constructed, which also gives a graphical construction developed by one of the author's assistants, Mr. J. F. Alcock, whereby the temperatures at the beginning and end of the expansion stroke can be read off with a fair degree of accuracy. Owing to the influence, previously mentioned, of pressure and charge proportions on dissociation the construction of such a curve to meet a reasonably wide range of conditions renders it necessary to compromise to some extent and to take mean values.

Thus, the energy curve shown applies strictly only to a benzene-air complete-combustion mixture at a compression ratio of 5 to 1. However, the effect of a change in the compression ratio is almost negligible within the limits used on the constant-volume cycle; as is the effect, also, of replacing benzene by any other hydrocarbon fuel. If used for a fuel such as alcohol or ether, however, the error becomes appreciable, owing to the different specific heat of the products of combustion. Strictly speaking, it does not apply to mixtures which are either weaker or richer than the chemically correct mixture, although here again the divergence is very small within the range available and with a homogeneous mixture.

The internal energy curve is plotted in terms of foot-pounds per standard cubic inch on a vertical scale, against the temperature on a horizontal scale. The other full-line curve shows the energy present as heat, so that the difference between the two curves shows the chemical energy stored in the products of dissociation. Zero energy is taken at 100° C. (212° F.), as being an average temperature at the beginning of compression. Variations in this temperature, being relatively small, will have but

[1] *Automobile Engineer*, February and March 1921.
[2] *Zeitschrift für physikalische Chemie*, 1912.

little influence on the combustion and exhaust temperatures. The explanation given below of the use of the diagram is supplemented

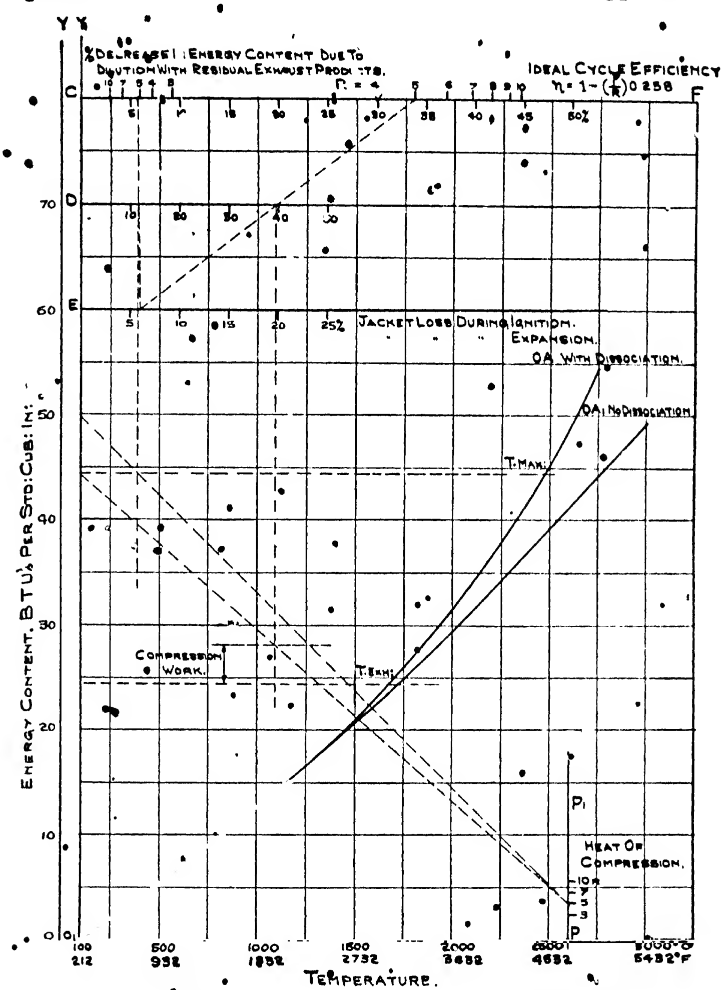


Fig. 17.—Internal Energy Diagram for working Fluid of Internal-Combustion Engine running on Volatile Hydrocarbon Fuels

by an example worked out for the following data, the construction lines of the example being shown dotted in the figure.

Compression ratio, R	=5 to 1	
Energy content	=46·2 ft. lb. per cub. in.	
Heat loss during combustion	=6 per cent	The actual value of these losses will be considered in Chap. III.
Heat loss during expansion	=6 per cent	

There are three factors in an actual engine which modify the temperature attained by the combustion of a mixture of any given energy content. They are the

(1) Heat put into the mixture by compression.

(2) Loss due to cooling by the walls of the combustion chamber during combustion.

(3) Effective weakening of the mixture due to dilution with the residual exhaust products.

Factor (1) is allowed for by laying off the heats of compression for various ratios by the marks "$R=5$," &c.; on the line PP_1 near the bottom of the diagram. The energy content is then marked off above this on the vertical line O_1Y_1 representing the 100° C. (212° F.) starting-point. In the example, the 46·2 ft. lb. energy content is laid off above the 3·6 ft. lb. of compression, making a total of 49·8 ft. lb., this being the gross energy content from which the losses due to factors (2) and (3) must be deducted. These are provided for in the following manner:—

On the horizontal scale C is marked the effective energy loss due to dilution with residual exhaust, assumed to be at 2100° F. abs. (=1165° C. abs.). Scale E shows the percentage loss due to cooling during combustion. This is laid off at any figure which previous experience shows as probable for the type of combustion chamber in question, namely, 6 per cent in the example. A line is then drawn between these two points, and the point of intersection of the line so drawn with the scale D gives the total percentage loss due to these two causes, corresponding to 11·5 per cent in the example chosen.

To transfer this to the diagram, a line is dropped vertically from the above intersection point. Another line is drawn from the point on the line O_1Y_1, giving the gross British thermal units per cubic inch, to the suitable compression point on the line PP_1, and representing 100 per cent on scale D; or 49·8 ft. lb. per cub. in. in the example. From the intersection of the above two lines, a horizontal line is run to the energy scale on one side, and to the energy curve on the other. The point on the energy scale shows the net energy available for expansion, or 44·5 ft. lb. per cub. in. in the example. From the energy-temperature curve the actual flame temperature can be read off, this is 2475° C. (4487° F.) in the example.

The drop in temperature during the expansion stroke depends on the two factors of (a) external work done, and (b) heat loss to the

walls. The net power output is given as a percentage of the heat content of the mixture on scale F, the formula used being $n = 1 - \left(\frac{1}{r}\right)^{0\cdot258}$. 0·258 is the mean value for the exponent, and covers all dissociation and similar effects, but not wall losses. The wall loss during expansion is laid off on scale E. A line drawn between these points gives their sum on scale D as before. A perpendicular from this point is dropped to meet a line from the net-energy point on O_1Y_1 to the suitable-compression point on the line PP_1. As the gross work done during expansion is the sum of the net work mentioned above and the compression work, this latter amount, which is 3·6 ft. lb. in the example, must be laid off below the above intersection point, to find the energy content at the end of expansion, which is 24·5 ft. lb. in the example. The corresponding final temperature, 1675° C. (3047° F.) in the example, can then be read off from the energy curve.

Taking an actual example from experiments on the author's variable-compression engine, with a correct mixture of an energy content of 46·2 ft. lb. per cub. in. and a compression ratio of 5 to 1, the actual maximum flame temperature, as obtained from the above diagram in fig. 17, allowing for the additional heat of compression, the wall loss during combustion and the dilution by residual exhaust products, will be 2475° C. (4487° F.), corresponding to an energy content of 44·5 ft. lb. per standard cub. in. At a ratio of 5 to 1, the observed indicated thermal efficiency is 31 per cent; of this, 5 per cent is due to the change in specific volume of the mixture, so that the heat drop is $46\cdot2 \times 31 \times 100/105 = 13\cdot6$ ft. lb. per cub. in. Add to this the 3·6 ft. lb. of compression work restored during expansion, and the 6 per cent of 46·2 or 2·8 ft. lb. of wall loss during expansion; the total heat-drop during expansion then becomes 20·0 ft. lb. per cub. in., leaving a final energy content of 24·5 ft. lb. per cub. in., which, it will be observed, coincides with the figure found in the example under the same conditions. The corresponding final temperature is 1675° C. (3047° F.)

While affecting the final temperature directly, it should be observed that the loss of heat during expansion has only a slight influence on the actual efficiency; this has been ignored in the construction, because much of it is lost late in the expansion stroke, where its value is less. Another slight error allowed to remain in the construction, for the sake of simplicity, is that a percentage of the *net* heat available during combustion is deducted for the jacket

loss during expansion; whereas this is given as a proportion of the *total* heat available in the fuel. The error due to this cause is, however, very small, being in the case considered 2·8 [(46·2 − 44·5) ÷ 44·5] = 0·11 ft. lb. per cub. in., and can be ignored safely.

At the end of the expansion stroke the temperature of the gases is about 3050° F. or 3509° F. absolute, as has been seen, and they are at an absolute pressure of about 70 lb. per square inch. When the exhaust valve opens these gases will expand rapidly down to atmospheric pressure, and their temperature will fall in the ratio $\left(\frac{14 \cdot 7}{70}\right)^{\frac{\gamma-1}{\gamma}}$, where γ is approximately 1·30. This brings their temperature down to 2450° F. abs., and the loss of heat during the exhaust stroke further reduces their temperature to about 2100° F. abs.

A similar calculation in the case of an ethyl-alcohol mixture makes the residual temperature about 1950° F. abs.

When the temperature at any point in the cycle has been found, the corresponding pressure is, of course, easily obtained. Thus, in the case of the petrol given above, the compression pressure is 123 lb. per square inch abs. and the temperature 1258° F. abs. As the temperature after ignition is 4487° F., or 4946° F. abs., and the ratio of the specific volumes before and after combustion 1·05, the pressure at the commencement of expansion is therefore $123 \times 1 \cdot 05 \times \frac{4946}{1258} = 508 \cdot 5$ lb. per square inch abs. or 493·8 lb. per square inch gauge.

At the end of the expansion stroke the volume has increased five times, and the temperature has dropped to 3509° F. abs. The pressure is therefore $\frac{123}{5} \times 1 \cdot 05 \times \frac{3509}{1258} = 72 \cdot 2$ lb. per square inch abs., or 57·5 lb. per square inch gauge.

The "explosion" pressure thus calculated is somewhat higher than that actually found in practice due to the finite speed of burning of the mixture, which causes a slight rounding of the peak of the diagram, though not enough appreciably to affect the thermal efficiency.

With benzene, the temperatures at the beginning and end of expansion will be 2470° C. (4480° F.) and 1670° C. (3040° F.) respectively, the corresponding pressures being 505 lb. per square inch abs. (490·3 lb. gauge) and 71·8 lb. per square inch abs. (57·1 lb. gauge) respectively.

From the above considerations we are able to construct an

indicator diagram giving the pressures and temperatures throughout the cycle under normal working conditions, such as would apply in the case of a small but efficient cylinder of the size quoted, and when running at a speed of 2000 R.P.M. The influence of cylinder size on the performance will be considered later in Chapter III, when it will be seen that it does not play any very important part; neither, indeed, within reasonable limits does the actual speed of revolution, provided always that it be fairly high, i.e. 1000 R.P.M. or over. At speeds below about 1000 R.P.M. and with a cylinder of the size under consideration, the loss of heat to the cylinder walls will begin to make itself felt, and will influence the performance appreciably.

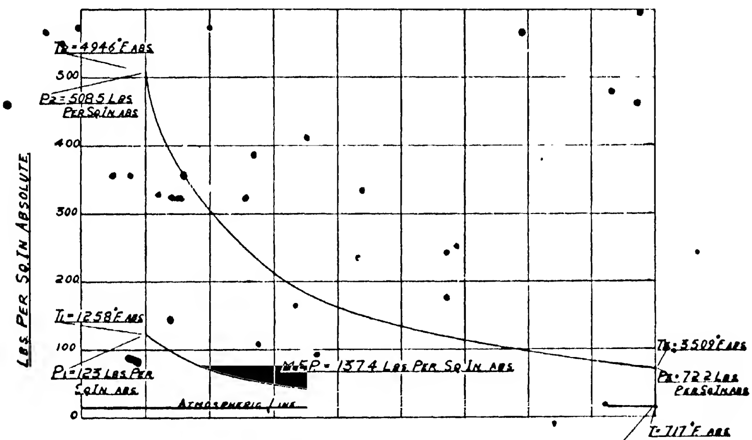


Fig. 18.—Indicator Diagram for "Correct" Petrol Mixture at Compression Ratio 5 : 1.

The indicator diagram shown in fig. 18 applies therefore strictly only to the conditions quoted above, though it would differ but little over a wide range of size or speed provided always that the combustion chamber were of an efficient form and that there is ample turbulence to spread combustion rapidly throughout the whole mass of the working fluid.

Once the thermal efficiency, the volumetric efficiency, and the total internal energy of the fuel-air mixture have been determined, it is possible to arrive also at the mean effective pressure and therefore the power output, by multiplying together the thermal efficiency, the total internal energy (in terms of ft. lb. per cubic inch × 12), and the volumetric efficiency.

In the case under consideration the thermal efficiency with a "correct" mixture, i.e. a mixture giving complete combustion without excess either of fuel or air, is 31 per cent. With a 20 per cent weak mixture it should be considerably higher on account of the lower maximum flame temperature, but this, in practice, is offset by the very much slower rate of burning, with the result that it will be increased only to about 32·5 per cent.

The energy liberated by the combustion of 1 standard cubic inch of petrol-air mixture was given as 46·2 ft. lb., but in so far as the available energy from the point of view of power output is concerned, we must multiply this by the change in specific volume, since, if the products of combustion occupy at the same temperature a greater volume than before, the pressures will be higher, though the temperatures remain unaltered. This factor has been taken into account in determining the maximum and terminal pressures, and it, of course, applies also to the mean effective pressure, which will be increased or decreased according as the specific volume increases or decreases; here petrol scores a marked advantage over benzene (benzol), and alcohol over either. It is convenient to use the term "total internal energy of the mixture" (as distinct from the heat energy liberated by combustion) to include the change in specific volume, i.e. the heat energy liberated, multiplied by the specific volume for a "correct" mixture. The total internal energy for the three fuels under consideration then becomes:

Fuel.	Change in Specific Volume.	Heat Energy liberated per Standard cub. in. ft. lb.	Total Internal Energy per Standard cub. in. ft. lb.
	per cent.		
Petrol	+5·0	46·2	48·5
Benzene	+1·3	46·9	47·5
Alcohol	+6·5	44·5	47·4

With petrol and a correct mixture in the example chosen, the thermal efficiency is 31 per cent, the internal energy 48·5 ft. lb. per standard cubic inch, and the volumetric efficiency 76·2 per cent. The indicated mean effective pressure will therefore be

$$·31 \times (48·5 \times 12) \times ·762 = 137·4 \text{ lb. per square inch.}$$

In the case of benzene the same calculation will give

$$·31 \times (47·5 \times 12) \times ·785 = 137·8 \text{ lb. per square inch,}$$

or, to all intents and purposes, exactly the same as with petrol, since the lower internal energy balances almost exactly the higher volumetric efficiency.

With alcohol, the same calculation will show that the indicated mean pressure should be 158·3 lb. per square inch, but this is based on a volumetric efficiency of 90 per cent, an efficiency which is not, in practice, realized for reasons already given. If the volumetric efficiency be taken as 82·5 per cent, which corresponds with experimental results under similar conditions, the indicated mean pressure becomes $158\cdot3 \times \frac{\cdot825}{\cdot900} = 145$ lb. per square inch, which is in very close agreement with observed results.

Again, for a 20 per cent weak mixture we must reduce the total internal energy accordingly, but, on the other hand, the thermal efficiency is increased owing to the lower flame temperature; also the volumetric efficiency is reduced very slightly owing to the smaller influence of the latent heat of evaporation for such a mixture. The figures then become:

Fuel.	Indicated Mean Pressure, lb. per sq. in.		Observed I.M.E.P. (Correct Mixture), lb. per sq. in.
	Correct Mixture.	20 per cent Weak.	
Petrol	137·1	118·0	136·0
Benzene	137·8	118·2	136·0
Alcohol	145·0	121·0	143·6

If, now, the mixture be further enriched by the addition of 20 per cent excess of fuel, although the heat liberated per standard cubic inch of mixture will be no greater, the total internal energy will be increased very slightly owing to the greater increase in specific volume when excess of fuel is present; again, owing to the greater latent heat available the volumetric efficiency will be increased slightly, while, finally, the thermal efficiency, based on that portion of the fuel which is burnt completely, will be slightly higher on account of the lower flame temperature; the net result of all these changes will be that the indicated mean pressure will, in all cases, except that of benzene, be some 4·5 per cent greater than with a correct mixture. In the case of benzene, however, owing to the small change in specific volume, the gain in power by using a

20 per cent rich mixture will be only about 3 per cent. If the mixture be enriched beyond 20 per cent excess of fuel; there will be no further gain in power on petrol or benzene owing to the slower rate of burning, but, on alcohol, there will still be a small further gain on account of its high latent heat and large increase of specific volume.

The indicated horse-power obtainable from the cylinder under consideration will be as follows :—

Fuel.	I.H.P.		
	20 per cent Weak.	Correct.	20 per cent Rich.
Petrol	15·75	18·35	19·15
Benzene	15·8	18·4	19·0
Alcohol	16·55	19·35	20·2

CHAPTER III

DISTRIBUTION OF HEAT IN A HIGH-SPEED FOUR-CYCLE ENGINE

It is usual to express the distribution of heat in an Internal-Combustion engine in terms of the proportion converted into indicated horse-power, the proportion yielded up to the cylinder walls, and, lastly, that rejected to the exhaust; the latter item being the residue after deducting from the total heat of the fuel the two former categories; it generally includes also the losses by radiation. This form of subdivision is perfectly legitimate so long as it is clearly understood that it is no more than a convenient form in which both to measure and to express the heat distribution; indeed, it is practically the only form in which it can be directly measured. Many engineers, however, still appear to be under the impression that it represents the true disposal of the available heat of the fuel, and, acting on this supposition, are often sadly misled.

The proportion of the total heat of the fuel converted to indicated horse-power can be determined readily enough and quite accurately from the known heat supplied and the known horse-power developed from it.

The heat yielded up to the cylinder walls and carried away by the cooling water can also be determined fairly accurately; it must be understood, however, that it includes:

(1) The heat given up by radiation, conduction, convection, &c., during the period of combustion.

(2) The heat given up during the expansion period.

(3) The heat given up during the exhaust stroke.

It is necessary to examine each of these sources separately.

(1) **Heat lost during Combustion.**—The period of combustion as distinct from expansion is relatively a short one, but it is one during which the ruling temperature in the combustion chamber is very high indeed, i.e. between 4200° F. and 4500° F. in the case of most volatile liquid fuels such as petrol, benzol, &c. Also it is

a period during which the gases within the combustion chamber are in a state of violent commotion so that heat is conveyed very readily by convection, &c.

Now, if by any means the loss of heat to the cylinder walls during this period could be suppressed, such heat could be converted into indicated horse-power at an efficiency corresponding to the efficiency of expansion alone (i.e. exclusive of the negative work done during compression), which in an engine with a compression ratio of 5 : 1 is roughly about 40 per cent. The remaining 60 per cent of the heat so recovered would, in any event, be rejected to the exhaust after expansion.

(2) **Heat lost during Expansion.**—Loss of heat during the expansion stroke may or may not be serious, depending upon the stage in the expansion stroke at which it is lost. Loss of heat at the very commencement of the expansion stroke is almost equally as serious as that lost during the combustion period, because had its loss been suppressed it would have been utilized at an efficiency corresponding to nearly the full ratio of expansion, whereas heat lost during the latter part of the expansion stroke is of very little moment, for even had its loss been suppressed it could have done but little useful work during the remainder of the expansion stroke and nearly the whole of it would have been rejected to the exhaust in any case.

At first sight it would appear that, owing to the higher temperatures and pressures ruling at the beginning of the expansion stroke, the loss of heat will be much greater during the earlier period, but against this it must be remembered that, as the expansion proceeds and the piston descends, an increasing area of cold cylinder barrel is exposed. Also, owing to dissociation and subsequent recombination, the fall in temperature during the expansion stroke is nothing like so great as it might appear ; the final temperature, even with a compression ratio of 5 : 1, being still well over 3000° F.

From the above considerations it will be seen that, though it is customary to yoke together the heat lost during combustion and expansion as though its influence during each period were the same, it is most certainly inaccurate and misleading to do so. Of the average heat loss during expansion, probably only about 20 per cent could be converted into useful work and the remaining 80 per cent would be rejected to the exhaust.

(3) **Heat lost during the Exhaust Stroke.**—Although during the exhaust stroke the temperature of the gases is very much lower, yet heat is given up to the cooling water with great rapidity during

this period, for in addition to the normal heat flow to the cylinder walls the hot gases are issuing at an exceedingly high velocity past the exhaust valve and through a short length of exhaust pipe which is always included in the cylinder jacket and cooled by the circulating water, consequently of the total heat carried away by the cooling water at least one-half and often more than half is given up during the exhaust period. Now the whole of the heat taken up during the exhaust stroke, by far the bulk of that taken up during expansion, and about 60 per cent of that taken up during combustion, should have been debited to the exhaust loss account.

In addition, of the total heat carried away by the cooling water, a very substantial proportion is generated by the friction of the piston on the cylinder walls.

It is interesting to take a specific example and to trace out as accurately as it is possible so to do the true gain in efficiency which would be effected if all heat loss to the cylinder walls were completely suppressed. Let us take, as a fair average example, the case of a well-designed and efficient engine with a compression ratio of 5 : 1 in which

32 per cent of the total heat of the fuel is converted into useful work on the piston,
28 per cent of the total heat of the fuel is carried away by the cooling water,
40 per cent of the total heat of the fuel remains and is accounted as lost to exhaust radiation, &c.

Of the total heat carried away by the cooling water, approximately 6 per cent will be lost to the walls of the cylinder during the combustion period, about 7 per cent will be yielded up during expansion, and the remaining 15 per cent during the exhaust stroke. Of the 6 per cent lost during the combustion period roughly about 40 per cent would appear as useful work or 2·4 per cent of the total heat of the fuel. Of the 7 per cent lost during expansion, somewhere about 20 per cent would be utilized or 1·4 per cent of the total heat of the fuel. Of the 15 per cent lost during the exhaust stroke, no part could have been utilized. We find therefore that, although 28 per cent of the total heat of the fuel has been carried away by the cooling water, only 3·8 per cent could have been converted directly into useful work on the piston, and the efficiency of the engine would be increased from 32 per cent to 35·8 per cent only, a gain of barely 12 per cent. Nor is this all, for, had all loss of heat to the cylinder walls been suppressed, the temperature of the working fluid would necessarily have been correspondingly higher, with the

result that the losses due to the increase both of specific heat and of dissociation at the higher temperatures would be increased substantially, and the net gain would be very small, probably only from 32 per cent to about 34·5 per cent or possibly 35 per cent.

These figures show clearly how relatively small a part the loss of heat to the cylinder walls plays in an Internal-Combustion engine, and how misleading it may be to assess that loss of heat by the heat carried away by the cooling water. As a first approximation it is fairly correct to assume that, of all the heat carried away by the cooling water of a cylinder, little more than 10 per cent could actually be converted directly into useful work.

Tables I., II., and III. show the distribution of heat as found in the author's $4\frac{1}{2}$ in. × 8 in. variable-compression research engine shown in figs. 3 and 4 under several different conditions. These figures were all obtained under circumstances which ensure a very high degree of accuracy.

Group A.—Distribution of heat at different speeds with wide open throttle. In all cases the mixture strength was approximately 10 per cent weak, the jacket temperature was kept constant at 140° F., and the heat input to the carburettor was maintained at 0·0433 B.Th.U. per revolution.

The three sets of tests in this group were run, one on ethyl alcohol 95 per cent, and the other on petrol, sample (A), at a compression ratio of 3·8 : 1, while the third set was run on ethyl alcohol 95 per cent at a compression ratio of 7 : 1.

It will be observed :

(1) That owing to the lower mean cycle temperature the thermal efficiency obtained with alcohol is substantially higher than that obtained with petrol at the same compression ratio.

(2) For the same reason, the proportion of heat carried away by the cooling water is less.

(3) That the thermal efficiency is affected very little by a wide variation in speed.

(4) That the proportion of the heat taken away by the cooling water falls slightly as the speed is increased.

The results shown in Group B were observed under the following conditions. The engine was run at a constant speed of 1500 R.P.M., corresponding to a piston speed of 2000 ft. per min., and the load was varied by throttling, the mixture strength being kept constant throughout at about 10 per cent weak, while the heat input to the carburettor was maintained at 0·0433 B.Th.U. at full load and

TABLE I—GROUP A

Fuel, Ethyl Alcohol 95 per cent Compression ratio 3·8 : 1				
R.P.M.	975	1300	1500	1700
Piston speed ft. per min. ...	1300	1733	2000	2266
Heat to I.H.P., per cent ...	26·9	27·0	26·9	27·0
„ cooling water, per cent	25·1	24·7	24·4	24·2
„ exhaust, radiation, &c.	48·0	48·3	48·7	48·8
Total heat	100·0	100·0	100·0	100·0
Fuel, Petrol, sample (A) Compression ratio 3·8 : 1				
R.P.M.	975	...	1500	1700
Piston speed, ft. per min. ...	1300	...	2000	2266
Heat to I.H.P., per cent ...	25·9	...	26·1	26·1
„ cooling water, per cent	30·4	...	28·0	27·0
„ exhaust, radiation, &c.	43·7	...	45·9	46·9
Total heat	100·0	...	100·0	100·0
Fuel, Ethyl Alcohol 95 per cent Compression ratio 7 : 1				
R.P.M.	975	1300	1500	.
Piston speed, ft. per min. ...	1300	1733	2000	.
Heat to I.H.P., per cent ...	37·6	38·1	38·3	.
„ cooling water, per cent	25·4	24·3	23·9	...
„ exhaust, radiation, &c., per cent	37·0	37·6	37·8	...
Total heat	100·0	100·0	100·0	..

reduced proportionately as the load was reduced. The circulating water temperature was maintained constant throughout at 140° F.

TABLE II—GROUP B

Fuel, Ethyl Alcohol 95 per cent Compression ratio 5·45 : 1 R.P.M. 1500				
Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	34·8	35·0	35·0	34·8
" cooling water, per cent	24·1	26·0	29·2	33·0
" exhaust, radiation, &c., per cent ...	41·1	39·0	35·8	32·2
Total heat	100·0	100·0	100·0	100·0
Fuel, Petrol, sample (A) Compression ratio 5·45 : 1 R.P.M. 1500				
Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33·5	34·0	34·1	33·5
" cooling water, per cent	26·5	28·2	31·8	35·5
" exhaust, radiation, &c., per cent	40·0	37·8	34·1	31·0
Total heat	100·0	100·0	100·0	100·0

From these tests it will be observed that:

(1) Provided that, as in the above cases, the time of ignition is advanced as the weight of charge is reduced, and therefore the relative proportion of residual exhaust products is increased, the thermal efficiency remains practically constant over a wide range of load.

(2) That as the load is reduced a greater proportion of the exhaust heat appears in the cooling water.

Lastly, a set of tests were run on hydrogen (Table III.). On this fuel alone it is possible to reduce the power output over a wide range

by controlling the supply of fuel only, i.e. by qualitative governing, without relying upon stratification. These tests are particularly interesting as they are, it is believed, the only experiments which have yet been carried out in which the power output has been varied solely by varying the mean mixture strength of the working fluid.

TABLE III

Fuel, Hydrogen Gas
Compression ratio 5·45 : 1
R.P.M. 1500
Mixture strength at maximum load 10 per cent weak

Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33·3	35·6	38·2	40·0
„ cooling water, per cent	23·6	24·9	25·3	28·6
„ exhaust, radiation, &c., per cent	43·1	39·5	36·5	31·4
Total heat	100·0	100·0	100·0	100·0

In these tests it will be observed that:

(1) The thermal efficiency increases rapidly as the load is reduced, due to the lower mean temperature.

(2) The proportionate heat supply to the cooling water increases slightly as the load is reduced, but at nothing like the rate shown in the previous tables, when governed quantitatively.

It is interesting to compare the results obtained on hydrogen with those found in the previous group. In the latter case the load is varied by varying the weight of working fluid per cycle, the temperature remaining practically constant, while in the tests with hydrogen the weight remained constant and the temperature was varied. As might be expected when the temperature is varied, so the efficiency follows suit, rising as the mean cycle temperature falls. Incidentally, also, the results obtained in Group B serve to illustrate how small a part direct heat loss to the cylinder walls actually plays, for as the weight of charge is reduced at nearly constant temperature, the *relative* heat loss must be increased substantially and yet the thermal efficiency remains unchanged. That quite a substantial change in heat loss has so little effect on the thermal efficiency illustrates how small, in itself, this whole source of loss must be—

that it actually effects no reduction at all in the thermal efficiency is probably due to the fact that as the weight of charge is reduced, so is the proportion of residual exhaust products increased, with the result that the temperature is slightly lower, and this very slight lowering of the flame temperature results in a gain in efficiency sufficient just to balance the increased relative heat loss.

The conclusions to be drawn from the preceding observations and tests are:

(1) That the direct loss of heat to the cylinder walls plays a comparatively insignificant part in the performance of an Internal-Combustion engine, and that even were the whole of this loss completely suppressed, the gain in power output and efficiency would be equivalent only to the conversion of an extra 2·5 to 3 per cent of the heat of the fuel into useful work.

(2) That, of the total heat carried away by the cooling water, only a very small proportion could be converted into useful work, and by far the bulk would appear in the exhaust.

(3) That when running with wide open throttle, the heat flow to the cooling water is very nearly proportional to the speed of the engine, i.e. the proportion remains nearly constant.

(4) When the load is reduced by throttling a much larger proportion of the heat rejected to the exhaust is taken up by the cooling water, before the gases enter the exhaust pipe.

It is frequently stated that the relative inefficiency of a combustion chamber with a large surface volume ratio is due to the increased heat losses. From the above it is clear that this cannot have any great influence. The most probable cause of the comparative inefficiency is, as shown in Chapter IV., that, owing to the lack of turbulence in such types of combustion chamber, an appreciable portion of the charge clings to and is so chilled by the walls that it escapes combustion altogether.

Influence of Cylinder Jacket Temperature.—It is frequently found that engines give more power and run more efficiently when the jacket water is hot. This is usually, but quite wrongly, attributed to the reduced heat loss to the cylinder walls when the jacket temperature is higher. It is due rather to the fact that the warming of the air and induction system is generally effected by means of the cylinder circulating water, consequently when the cylinder jackets are cold, so also is the induction system, with the result that a considerable proportion of the fuel precipitates, the distribution becomes defective, and both the power and efficiency suffer in conse-

quence. It is due also to the fact that the piston friction is dependent upon the viscosity of the oil on the cylinder walls and this, in turn, is dependent upon their temperature.

In engines in which the temperature of the induction system is controlled independently of the cylinder temperature, the difference in power, if any, is much less marked, and it then becomes a question of the relation between three factors.

(1) The variation in heat loss to the jackets with temperature.

(2) The variation in volumetric efficiency with temperature.

(3) The variation in piston friction with temperature.

With regard to the heat loss to the jackets. It has already been shown that with a reasonably well-designed combustion chamber the whole of the heat flow to the cooling water during combustion and expansion amounts to only about 12 to 13 per cent of the total heat of the fuel, and that if the whole of the loss during combustion and expansion were completely suppressed, the indicated horsepower would be increased by less than 10 per cent, taking into account the loss due to dissociation, &c., at the higher temperatures, which would then be reached. Now the mean temperature during combustion and expansion may be taken as about 3800° F., and that of the inner surface of the cylinder walls with boiling jacket water as at most about 300° F., the temperature difference between the two being about 3500° F. If now the temperature of the jacket water be reduced by 150° F. (that is, from boiling to 72° F.) the temperature difference between the gases and the cylinder walls will only be increased to 3650° F., an increase of little more than 4 per cent. Assuming that the heat loss is proportional to the temperature difference (and this is approximately true), then the difference in the indicated efficiency and power due to the greater heat loss with cold jackets will only be 4 per cent of 10 per cent, or about 0·4 per cent. With a very badly designed combustion chamber it might conceivably amount to as much as 1 per cent, but in any case it is but a trifling amount.

With regard to the variation in volumetric efficiency the difference here is much more marked and is in the reverse direction. With hot cylinder walls at 300° F., the rise in temperature of the working fluid due to contact with the inlet valve and cylinder walls during its entry will amount to about 80° F. With cold water it will be about 55° F., assuming that the mean surface temperature of the walls is, in each case, about 90° F. higher than that of the jacket water. Experiments quoted elsewhere show that with a cylinder of

normal design the change in the temperature rise of the entering gases is about one-sixth of that of the cylinder walls. The mean absolute temperature of the working fluid after its entry to the cylinder may be taken as about 700° F. absolute. The weight of charge taken into the cylinder and therefore the power output will be proportional to the absolute temperature, so that if this is reduced by 25° F. the weight of charge taken into the cylinder, per cycle, will be increased in the proportion of $\frac{700}{675}$, or about 3·75 per cent. From the above considerations it will be seen that while the power and efficiency

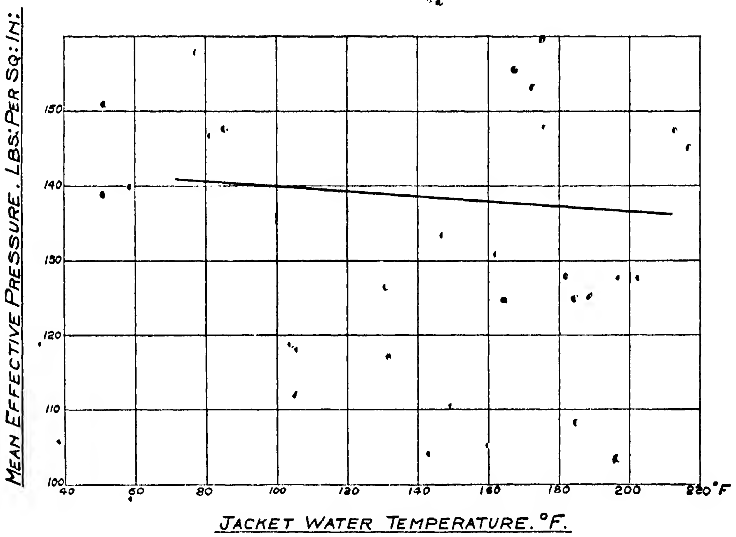


Fig. 19.—Variation in indicated Mean Pressure with Cylinder Temperature

may be reduced by from 0·4 per cent to 1 per cent, due to increased heat losses, the weight of charge taken in will be increased by about 3·75 per cent, the net result being a gain of about 2·75 to 3·25 per cent in the indicated horse-power by reducing the temperature of the cooling water from boiling to about 72° F. or what is usually termed stone cold.

So far as the indicated horse-power is concerned, therefore, any reduction in the temperature of the cooling water involves an appreciable increase in the power developed in the cylinder. Fig. 19 shows the mean results of a large number of actual tests, from which it will be seen that the observed results agree very closely with the above deductions.

The third factor, namely piston friction, plays a very important part. In Chapter IX, dealing with piston design, it will be emphasized that piston friction is dependent in a very large measure on the viscosity of the lubricating oil, and therefore upon the temperature of the cylinder walls—the colder the walls, the higher the viscosity of the oil clinging to them, and therefore the greater the friction. In the case of pistons with large bearing surfaces and heavy inertia pressures the difference in friction between hot and cold jackets may amount to as much as 8 per cent of the indicated horse-power of the engine, in which case the gain in indicated horse-power from the use of cold circulating water would be completely swamped by the extra friction, and the net result would be a loss of about 5 per cent in brake horse-power. On the other hand, with a very light piston, with the minimum of bearing surface, the difference may be as little as 3 per cent, in which event the brake horse-power of the engine will be independent of the jacket water temperature. Quite recently tests have been carried out on three single-cylinder engines, each of about the same size, in which the carburettor and induction system were heated independently of the circulating water. One of these engines was fitted with a cast-iron trunk piston of orthodox design, one with a very light aluminium slipper-type piston, and one with a cross-head piston, in which the friction of the piston on the cylinder walls is practically negligible. The results obtained are given below :

	B.H.P. at 1200 R.P.M.			
	Engine No.	Jacket Water, 212° F.	Jacket Water, 80° F.	Difference per cent.
(1) Cast-iron trunk piston ...	B. 8	27	25·7	−5
(2) Aluminium slipper-type piston	E. 35	28·3	28·3	Nil
(3) Cross-head piston	T.S. 1	29·4	30·0	+2

In the case of the engines used for tanks which were fitted with cross-head pistons and independently heated induction systems, it was invariably found that the brake horse-power was from 1·5 per cent to 2 per cent higher with "cold" jacket water, i.e. about 80° F., than with "hot," i.e. about 180°-200° F. The difference in power was quite appreciable in service, and tank drivers frequently expressed

surprise that their engines appeared to be more "lively" when the cylinders were cold.

To sum up, apart from any question of carburettor temperature, the power output of an engine may increase or decrease as the jacket temperature is raised, depending upon the piston friction; if the piston friction is high, the power output will increase; if low, it will not increase. The increase in heat loss over the most extreme conditions is so small as to be negligible. The increase in volumetric efficiency is comparatively large, but is usually not sufficient to balance the increase in piston friction, hence the loss of power so often observed.

In most cases, however, the temperature of the carburettor and induction system is dependent also upon the jacket-water temperature, and when operating with fuels of low volatility the variation of temperature of these parts may play a supremely important part, particularly if the distribution is inherently defective or the mixture is rather weak.

Gas Velocity and Indicated Mean Pressure.—In the previous volume and in various publications, &c., the author has assumed that the best all-round compromise between such various conflicting conditions as (1) the attainment of the highest possible volumetric efficiency, (2) the attainment of the necessary degree of turbulence, (3) the reduction of the fluid pumping losses to the lowest possible limit, is achieved when the mean gas velocity through the valves is in the neighbourhood of 130 ft. per second. Clearly from the point of view of conditions (1) and (3) it is desirable to keep the velocity as low as possible, while to fulfil condition (2) a high gas velocity is required. The figure quoted for the gas velocity was arrived at after a careful analysis of a very large number of published tests, from which it was found that the points of maximum torque and maximum efficiency both fell between the limits of gas velocity of 120 and 140 ft. per second. The gas velocity, as is customary, is expressed in terms of feet per second through the valve opening on the assumption (*a*) that the valve is wide open throughout the entire stroke; (*b*) that the mean and not the maximum piston speed is taken into account.

There is now abundant evidence that where the valves open directly into the main body of the combustion chamber a somewhat higher gas velocity may be employed without impairing the volumetric efficiency; on the other hand, with such a combustion chamber free from pockets or recesses turbulence will be better

maintained and a lower gas velocity will suffice. In other words, when the valves are so placed that they open freely into the combustion space, both the efficiency and the mean pressure are less dependent upon gas velocity. In practice, however, the size of the valves and valve passages is largely controlled by mechanical limitations. Practical experience indicates that, when valves are fitted directly in the cylinder heads, only as much area of opening should be given as can be provided without allowing the symmetry and compactness of the combustion chamber to suffer. It would seem preferable, therefore, to increase the gas velocity up to as high as 160 ft. per second, rather than to distort the combustion chamber in order to accommodate larger valves.

The valve-opening diagram is clearly a very important factor, and in the absence of precise information it must be assumed that the valve-opening diagram is in each case that best suited to the general design of the engine and its pipe-work.

The author has now a great deal more information on this subject, derived from a very large number of engines tested in his own laboratory and under his own observation. Further, since all these engines were designed on the same basis having the same valve-opening diagrams, and all running at sufficiently high speeds to eliminate any appreciable variation in the heat losses, the results are directly comparable; and the deductions to be drawn from them are probably quite reliable. The results of some of these tests are shown in fig. 20, from which it will be seen that, while both the mean pressure and fuel consumption differ very considerably, depending both upon the compression ratio and the efficiency of the combustion chamber, the relation between mean pressure, efficiency, and gas velocity is substantially the same in all cases.

Again, a series of experiments was carried out on the special single-cylinder research engine illustrated in figs. 3 and 4; this engine is fitted with two inlet and three exhaust valves. Provision is made for throwing any of the valves out of operation. The results obtained with various combinations of valves in action are shown in fig. 21.

Many useful deductions can be drawn from the results of this last series, the more so because: (1) The tests were carried out under conditions ensuring an exceptional degree of accuracy; (2) the area of valve opening, and therefore the gas velocity, could be altered without disturbing any other condition.

All experience serves to indicate that the velocity through the

exhaust valves may be as much as 50 per cent higher than through the inlet valves without causing any appreciable resistance or other loss. Although the same weight of working fluid has to pass through both the inlet and exhaust valves in much the same time, the conditions under which it passes are very different. In the first place, at the time when the exhaust valves open, the pressure in the cylinder is relatively high, generally from 50 to 70 lb. per square inch above atmosphere. The sudden release of the gases at this high pressure results in the setting up in the exhaust pipe of a very high

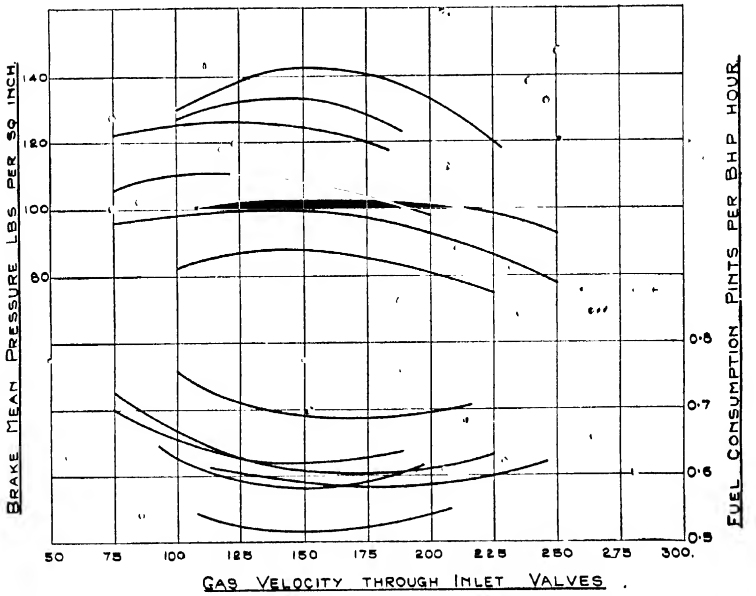


Fig. 20.—A Selection of Curves showing M.E.P. and Fuel Consumption on a Gas-Velocity Basis

velocity, and the kinetic energy acquired by the gases in this pipe assists greatly in the withdrawal of the remainder from the cylinder. Unless, therefore, the silencer offers any undue resistance the energy expended in driving out the exhaust products is almost negligible, and, within limits, is almost independent of the gas velocity.

While distinct evidence of wire-drawing accompanied by a rapid increase in the fluid pumping losses becomes apparent so soon as the inlet gas velocity exceeds about 150 to 160 ft. per second, there is no evidence of any measurable fluid resistance, at all events until the exhaust gas velocity exceeds about 240 ft. per second.

One reason for this is that owing to their high release pressure

the bulk of the exhaust products is discharged while the piston is more or less at rest. It must also be remembered that while a back pressure of, say, 1 lb./in. on the exhaust stroke only decreases the effective mean pressure by that amount, a negative pressure of 1 lb./in. at the end of the suction stroke will reduce the mean pressure by $\frac{1}{14 \cdot 7}$ or nearly 7 per cent, a much greater loss.

From the more reliable data now available it is possible to construct a curve giving the relation between inlet gas velocity and volumetric efficiency with a very fair degree of accuracy. As already explained, however, the volumetric efficiency is also influenced to a very large extent by the vaporisation of fuel within the cylinder and upon the degree of pre-heating before entry; it is therefore dependent upon these factors also. The curve shown in fig. 22 gives to a close approximation the relation between the volumetric

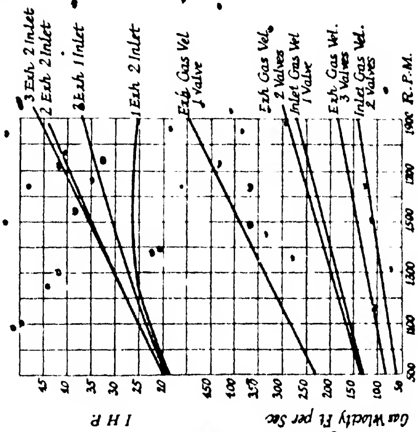


Fig. 21

TESTS WITH ONE OR MORE VALVES THROWN OUT OF OPERATION

I.H.P. (Compression ratio 6:1 in all cases.)

Number of Valves in Operation and Mean Gas Velocity.

	Column No. 1			Column No. 2			Column No. 3.			Column No. 4		
	No. of Valves 2 inl. 3 exh.			No. of Valves 2 inl. 2 exh.			No. of valves 2 inl. 1 exh.			No. of valves 1 inl. 3 exh.		
	Gas velocity, ft. per sec.		I.H.P.	Gas velocity, ft. per sec.		I.H.P.	Gas velocity, ft. per sec.		I.H.P.	Gas velocity, ft. per sec.		I.H.P.
R.P.M.	Inlet.	Exhaust.		Inlet.	Exhaust.		Inlet.	Exhaust.		Inlet.	Exhaust.	
900	66·1	86·6	19·9	66·1	137·2	20·0	66·1	235·0	18·8	132·2	86·5	19·1
1100	81·0	106·0	24·5	81·0	168·0	24·55	81·0	287·5	22·9	162·0	106·0	23·0
1300	95·5	125·0	29·4	95·5	198·0	29·4	95·5	339·0	26·0	191·0	125·0	27·0
1500	111·0	144·5	34·5	111·0	228·5	34·4	111·0	394·0	26·7	222·0	144·5	30·7
1700	124·8	163·0	39·8	124·8	259·0	39·3	124·8	443·0	26·4	249·6	163·0	33·8
1900	139·6	183·0	45·1	139·6	290·0	44·3	139·6	496·0	25·2	279·2	183·0	36·0

efficiency obtained and inlet gas velocity for an efficient and up-to-date design of petrol engine with normal valve timing; for an engine running on benzol it is about 2·5 per cent too low, and for alcohol about 8 per cent too low, provided, of course, that the amount of pre-heating is the same for each fuel. The curve also assumes that the induction piping and carburettor offer no abnormal resistance, and that not more than four cylinders draw their supply from any one carburettor.

To sum up, all the evidence available to-day tends to confirm the assumptions made in the first volume that the most satisfactory

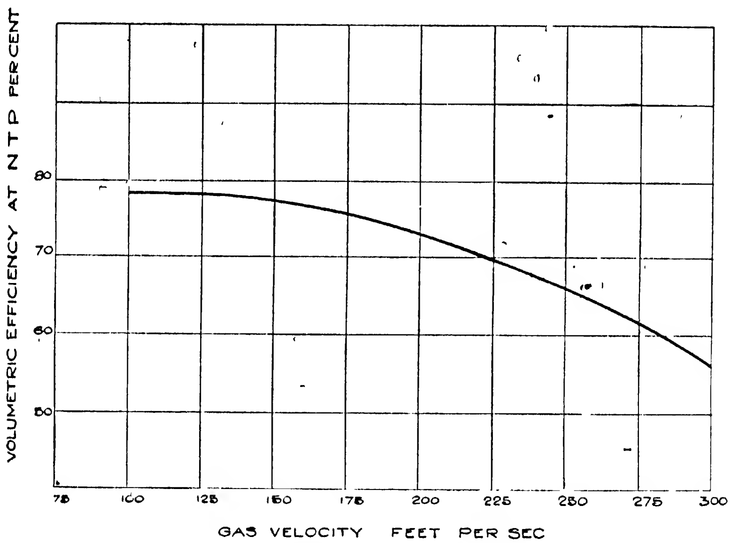


Fig. 22.—Volumetric Efficiency and Gas Velocity

compromise is attained when the gas velocity through the inlet valves is from 130 to 160 ft. per second, depending upon their position in the combustion chamber; the gas velocity through the exhaust valves may, however, be as much as 50 per cent greater without any ill effect.

Influence of Cylinder Size on Power and Efficiency.—For equal compression ratios the indicated power and efficiency of an engine is influenced by two factors: (1) Incomplete combustion of the layer of working fluid which clings to the walls of the combustion chamber; (2) direct loss of heat to the walls of the combustion chamber.

Both depend clearly upon the relation between the surface of

the combustion chamber and its volume, but the former depends also upon the thickness of the layer adhering to the walls, which in turn is dependent upon the degree of turbulence or scouring effect. Both these sources of loss are reduced to a minimum when the combustion chamber is as compact and symmetrical as possible, for not only is the surface volume ratio then reduced to a minimum; but the scouring effect of turbulence has the fullest possible play.

Assuming that the combustion chamber is similar in all cases, then it is evident that the relation of surface to volume will be reduced as the size of the cylinder is increased; since, so long as the proportions are similar, one factor varies as the square of the size and the other as the cube. It follows, therefore, that under similar conditions the power output per unit volume and efficiency of an engine will increase with increase of size. Exactly how much it will increase must depend upon the proportion which these sources of loss bear to the whole, that is to say, the gain in power and efficiency with larger cylinders must depend largely upon the design of the combustion chamber. When the combustion chamber in all cases is as compact and symmetrical as possible, the gain with increase of size will be at a minimum.

Actual values for the increase in power with cylinder size can only be arrived at, apart from theoretical reasoning, by comparing engines of similar design in every respect. To attempt to make deductions, as has so often been done in the past, from comparisons of engines of different designs and by different makers is quite useless, because so many other variables are necessarily introduced, as completely to mask the relatively small differences looked for. Also comparisons of engines by one designer are of little value unless all the engines compared are of fairly high efficiency; many misleading deductions have even been drawn from comparative tests of engines of similar design, but in which the efficiency and power output was so low as to indicate the presence of some other large factor—probably a variable one which has confused the issue.

Some years ago the Gas Engine Research Committee of the Institute of Civil Engineers carried out a very careful investigation upon three substantially similar single-cylinder gas-engines of 6, 24, and 60 indicated horse-power respectively, all of which were supplied by the National Gas Engine Co. and all of which were of the same general design. Of these three engines the indicated thermal efficiency was found to be 31·8 per cent, 33·3 per cent, and 34·7 per cent, and the indicated mean effective pressure 80, 88·7, and 86·2 lb.

per square inch respectively. It will be seen that while the indicated thermal efficiency increases with increase of cylinder size in regular steps, the increase in mean pressure does not. No explanation is given as to why this should be, and in the absence of any such explanation it must be presumed either that the volumetric efficiency of the three engines is not the same—or that a weaker mixture was used in the largest engines, though this would vitiate the comparative efficiency figures. It should be noted, however, (1) that all three engines ran at speeds which, at the present day, would be considered very low, consequently the proportion of heat loss would be somewhat higher than in modern high-speed engines; (2) that the combustion chambers were not as compact as they might have been, and therefore again the efficiency in all three cases is rather low, when the very favourable nature of the working fluid with its wide range of burning and low-flame temperature, as compared with petrol, are taken into account.

The heat balance sheet for the three engines after analysis and correction by Sir Dugald Clerk is as follows :—

Designation of Engine.	L.	R.	
Heat to indicated horse-power	31·8	33·3	34·7
Heat to cooling water	34·1	29·6	25·4
Heat to exhaust, radiation, &c.	34·1	37·1	39·9
Total heat	100·0	100·0	100·0

From this balance sheet it will be seen that the heat carried away by the cooling water diminishes as the cylinder size is increased. The figures given above represent the total heat carried away by the cooling water and by radiation, and not the loss of heat during combustion and expansion, which alone can affect the power output and efficiency. In the case of the X engine Sir Dugald Clerk was able to deduce the heat loss during combustion and expansion, and found that it amounted to 16·1 per cent or 63·3 per cent of the total heat carried away by the cooling water. If we take the same proportion as applying to the L and R engines (and in the absence of more accurate information this is as near the truth as it is possible to approach), then we find that the heat loss during combustion and expansion in the case of the L and R engines is 21·6 per cent and

18·7 per cent respectively. The corrected heat balance for the three engines then becomes.

Designation of Engine.	L.	R.	X
Heat to indicated horse-power	31·8	33·3	34·7
Heat loss during combustion and expansion	21·6	18·7	16·1
Heat to exhaust, radiation, &c.	46·6	48·0	49·2
Total heat	100·0	100·0	100·0

and the true heat loss apparently ranges from 21·6 per cent in the case of the smallest to 16·1 per cent in the case of the largest cylinder. The ideal efficiency for all three engines as calculated by the Committee from the specific heat of the working fluid actually used is 39·5 per cent, so that the relative efficiency of the three engines becomes 80·5 per cent, 84·3 per cent, and 87·8 per cent respectively.

The above tests are probably by far the most accurate and representative tests to determine the variation in performance with cylinder size that have yet been undertaken, but the information obtained from them is not altogether applicable to high-speed engines using rich volatile liquid fuels, because (1) the speeds are very low; (2) the flame temperature is very much lower owing to the presence of inert gases in the fuel and to the relatively wide range of burning on the weak side.

Quite recently the author has carried out a somewhat similar series of tests on three engines of similar design. These engines may be designated as A, B, and C. They all have the following characteristics:—

(1) The mechanical features are the same in all cases in so far as they can apply over so wide a range of size. In all three engines aluminium slipper-type pistons are used.

(2) In all cases the combustion chamber is as compact and symmetrical as is possible consistent with the provision of adequate valve area, and in all cases the valves are fitted in the cylinder head.

(3) In all three engines the valve opening diagram is identical.

(4) In all three the compression ratio is the same, namely 4·84.

The stroke-bore ratio unfortunately differs very widely. With engines, however, in which the combustion chamber is very compact

and free from valve pockets, etc., and the compression ratio fairly low, the stroke-bore ratio appears to play very little part.

The leading particulars of the three engines are as follows :—

Designation of Engine.	A.	B.	C.
Cylinder bore, inches	3·25	4·5	8·0
Stroke, inches	4·0	8·0	11·0
Swept volume, cubic inches	33·2	128	554
Speed at which maximum torque and maximum efficiency are developed, R.P.M.	1750	1750	1250
Volume of clearance space, cubic inches ...	8·65	33·4	144
Area of surface exposed to combustion, square inches	29·5	71·0	151
Ratio of surface to volume	3·41 : 1	2·12 : 1	1·048 : 1

Engines A and B were tested in the author's laboratory at Shoreham, and C at the Royal Aircraft Establishment at Farnborough. All three engines have been in operation since the latter part of 1919, and have been submitted to a great number of calibration tests by different observers. The fuel used in all three cases was a light petrol having a lower calorific value of 18,900 B.Th.U. per lb.

Engine A has two valves, one inlet and one exhaust, mounted vertically in the cylinder head.

Engine B has five valves, two inlet and three exhaust, all mounted vertically in the head.

Engine C has four valves, two inlet and two exhaust, also mounted in the cylinder head, but slightly inclined from the vertical.

Both the time of opening and closing and the gas velocity through the valves is the same in all cases at the speeds stated, namely 140 ft. per second. All three engines are designed to operate over a wide range of speed, particularly engine B, and all were tested with a mixture strength approximately 5 per cent weak in each case. Only in the case of engine B could the exact mixture strength be determined by simultaneous measurements of the air and fuel, but by applying the curve showing the relationship between mixture strength and M.E.P. obtained from engine B to engines A and C, in both of which the mixture strength was varied over a wide range, it is possible to determine with reasonable accuracy the corresponding mixture strength in each test on the other two engines.

The engine speeds corresponding to a gas velocity of 140 ft. per second through the valves are:

	A.	B.	C.
R.P.M.	1750	1750	1250
Piston speed, ft. per min.	1166	2333	2290

All three engines were tested without any pre-heating to the carburettors or induction system.

The mean results from a large number of tests are given in the following table :—

Engine Designation.	A.	B.	C.
B.H.P.	8·48	35·1	118
R.P.M.	1750	1750	1250
Mechanical efficiency, per cent ...	83	86	88
Brake M.E.P.	114	126	135
Indicated M.E.P.	137·5	146·5	153
Fuel consumption, lb. per B.H.P. hour	0·55	0·510	0·480
Fuel consumption, lb. per I.H.P. hour	0·456	0·438	0·422
Brake thermal efficiency	21·6 %	26·1 %	28·1 %
Indicated thermal efficiency	29·6 %	30·8 %	32·0 %

It should be noted that the above figures are taken from test results when all three engines were operating with a mixture strength containing about 5 per cent excess of air. A slightly higher thermal efficiency could be obtained in each case by weakening the mixture a further 10 per cent and a somewhat higher power by enriching it. The ideal efficiency for the three engines for a mixture strength 5 per cent weak and for a compression ratio of 4·84 is approximately 36·5 per cent, so that the relative efficiencies become 81·0 per cent, 84·5 per cent, and 87·7 per cent respectively.

At these revolutions the mechanical efficiency of the three engines is 83 per cent, 86 per cent, and 88 per cent respectively. The brake mean effective pressure is in the three cases 114, 126, and 135 lb. per square inch respectively. The corresponding indicated mean pressure is 137·5, 146·5, and 153·0 lb. per square inch.

Since the valve-opening diagrams and the mean gas velocity were the same in all cases, it may be presumed that the volumetric efficiency was also very nearly the same.

Fig. 23 shows the brake and indicated thermal efficiency and the brake and indicated mean effective pressure in terms of cubic inches of piston displacement per minute for the three engines. It is interesting to note that the relative results are in fairly close agreement with those obtained by the Gas-Engine Research Committee from three slow-running gas-engines, though they cover a somewhat wider range of cylinder size. In the case of the high-speed petrol

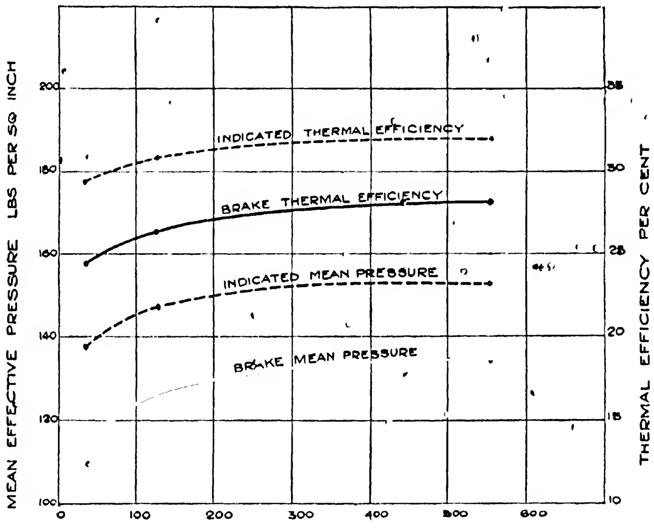


Fig. 23

engines, the compression ratio is considerably lower and the flame temperature very much higher, as is obvious from a comparison of the mean pressures, for despite the lower compression the indicated mean pressure of the C engine is 153 lb. per square inch, or very nearly double that of the X engine tested by the Gas-Engine Research Committee. In spite, however, of the much higher temperatures, the proportionate heat flow to the cylinder jacket in the case of these small high-speed engines is actually lower than in the gas-engines, while the ratio between the actual and the ideal efficiency is substantially the same.

Heat balance tests were taken from engines A and B, but not

The engine speeds corresponding to a gas velocity of 140 ft. per second through the valves are:

	A.	B.	C.
R.P.M.	1750	1750	1250
Piston speed, ft. per min.	1166	2333	2290

All three engines were tested without any pre-heating to the carburettors or induction system.

The mean results from a large number of tests are given in the following table:—

Engine Designation.	A.	B.	C.
B.H.P.	8·48	35·4	118
R.P.M.	1750	1750	1250
Mechanical efficiency, per cent ...	83	86	88
Brake M.E.P.	114	126	135
Indicated M.E.P.	137·5	146·5	153
Fuel consumption, lb. per B.H.P. hour	0·55	0·510	0·480
Fuel consumption, lb. per I.H.P. hour	0·456	0·438	0·422
Brake thermal efficiency	21·6 %	26·1 %	28·1 %
Indicated thermal efficiency	29·6 %	30·8 %	32·0 %

It should be noted that the above figures are taken from test results when all three engines were operating with a mixture strength containing about 5 per cent excess of air. A slightly higher thermal efficiency could be obtained in each case by weakening the mixture a further 10 per cent and a somewhat higher power by enriching it. The ideal efficiency for the three engines for a mixture strength 5 per cent weak and for a compression ratio of 4·84 is approximately 36·5 per cent, so that the relative efficiencies become 81·0 per cent, 84·5 per cent, and 87·7 per cent respectively.

At these revolutions the mechanical efficiency of the three engines is 83 per cent, 86 per cent, and 88 per cent respectively. The brake mean effective pressure is in the three cases 114, 126, and 135 lb. per square inch respectively. The corresponding indicated mean pressure is 137·5, 146·5, and 153·0 lb. per square inch.

CHAPTER IV

INFLUENCE OF FORM OF COMBUSTION CHAMBER

Of all the features of design which control both the power output and efficiency of an internal combustion engine, by far the most important is the form of the combustion chamber. Upon this depends not only the efficiency with which the fuel is burnt, and therefore the power output and efficiency of the engine, but also, to a very large extent, the liability to detonation; and detonation by limiting the compression ratio that may be used sets an additional limit on the efficiency.

In the design of the combustion chamber the most important considerations are:

(1) The maintenance of the turbulence set up by the gases during their entry;

(2) The position of the ignition plug;

(3) The avoidance of any pockets where the gases may become stagnant;

(4) The provision of a free and unobstructed entry for the gases after passing through the inlet valve.

Turbulence.– The maintenance of turbulence until the time of ignition is probably the most important consideration of all, for upon turbulence depends the rate at which combustion takes place. If the combustible mixture were completely stagnant at the time of ignition, the initial flame brought into life by the passage of the spark across the points of the ignition plug would spread so slowly that, even in a comparatively slow-speed engine, barely half the working fluid would be burnt before the exhaust valve opens. No matter what fuel be used (except possibly hydrogen), we have to rely almost entirely on turbulence or mechanical disturbance to distribute the wan and timid flame rapidly throughout the combustible mixture, and this becomes the more important as the density of the charge is reduced by throttling; for as the density is reduced so is the

proportion of residual exhaust products increased, and these, being diluents, tend to lower the flame temperature and so greatly to discourage and retard the process of inflammation. We rely, therefore, entirely upon mechanical disturbance, or turbulence, to speed up the process of combustion and to spread inflammation rapidly throughout the whole bulk of the working fluid.

Apart, however, from the question of accelerating combustion, turbulence plays in all probability another and highly important part. In any internal-combustion engine cylinder there is always a layer of working fluid adhering more or less to the cold cylinder walls. This, by its close proximity to cool surfaces, can get rid of its heat so rapidly that it escapes complete combustion. It is upon turbulence that we have to rely, to scour away this layer and to distribute it throughout the combustion chamber. There is little doubt but that the effective thickness of this layer, and its influence upon the power output and efficiency of an engine, depend largely upon the degree of turbulence within the combustion chamber.

Turbulence in itself, so far as the author has been able to ascertain, has little or no direct influence upon the tendency of the combustible mixture to detonate. If the mixture is completely stagnant and ignited from any one point, the flame will extend from that point very slowly at first, but at a steadily increasing velocity until ultimately it reaches a speed at which the unburnt gas, compressed by the rapidly approaching flame, receives heat from compression, and also by radiation and conduction at a rate in excess of that at which it can get rid of it to the cool cylinder walls, with the result that its temperature is raised above its self-ignition temperature. It then ignites spontaneously and almost instantaneously throughout its whole bulk, thus setting up an explosion or detonation wave. If instead of being stagnant the combustible mixture is in a state of violent disturbance at the time when ignition is started, the whole process is speeded up enormously; but it is the *whole* process that is so speeded up and not any one part of it, with the result that the phenomena are precisely the same, though on a different scale so far as speed is concerned.

It is generally supposed that a combustion chamber having a large surface in relation to its volume owes its low efficiency entirely to direct heat loss to the cylinder walls, but in Chapter III. we have seen that direct heat loss plays a relatively small part in the performance of an engine, and it is far more probable that this form of combustion chamber owes its low efficiency rather to the fact

that not only is the area of the layer of gas which clings to the walls and so escapes combustion considerably greater, but, in addition, its thickness also is greater, due to the fact that in such a chamber there is more resistance to the motion of the gases, and the scouring effect of turbulence is therefore less. Although it is not possible to state definitely that this is the case, yet there is a great deal of presumptive evidence in favour of such an assumption; not the least of which being that in any given design of combustion chamber an increase in turbulence, e.g. by increasing the velocity through the inlet valve, will often produce an increase in efficiency without altering in any respect the flow of heat to the cooling water. This, in itself, would not be conclusive evidence, but coupled with a great deal of other indirect evidence it carries a good deal of weight.

Position of the Ignition Plug.—The position of the ignition plug in the combustion chamber is very important, for upon it depends:

(1) The tendency to detonate;

(2) The time taken to complete combustion.

When dealing with the subject of detonation, it has been shown that the process of inflammation spreads from the point of ignition, slowly at first, but with a rapidly increasing velocity, and that detonation occurs when this velocity exceeds a certain rate in relation to the flame temperature. It is clear, therefore, that the greater the distance that the flame has to travel before reaching the farthest corner in the combustion chamber, the greater will be the tendency to detonate, and that, in order to reduce detonation to a minimum, the combustion chamber should be compact with the ignition plug as nearly in the centre as possible. Apart from the question of distance alone, the tendency to detonate depends also, in all probability, upon the temperature of the surfaces against which the unburnt gases are driven by the burning portion, because this governs largely the rate at which they can get rid of their heat.

From this argument it would appear evident that, for an equal tendency to detonate, the distance from the point of ignition to a hot surface should be less than that to a cool surface. The hottest surface in any combustion chamber is the head of the exhaust valve. It follows, therefore, that in order to reduce the tendency to detonate to a minimum the ignition plug should be placed not quite in the centre of the combustion chamber, but with a bias towards the exhaust valve head.

As illustrating the importance of the position of the ignition

plug in relation to the tendency to detonate, the author would quote two extreme examples which he has examined. In one the engine had a T-head combustion chamber with the inlet and exhaust valves on either side of the cylinder and with the ignition plug placed vertically over the inlet valve, so that inflammation spread from the ignition plug across the head of the cylinder and so into a shallow pocket, bounded on one side by the hot exhaust valve head and on the other by an uncooled cap for the removal of this valve. This engine detonated very heavily, and could only be run at full power on petrols containing a large proportion of aromatics, despite the fact that its compression ratio was only 3·9 : 1. Its power output and efficiency were very poor in consequence both of the low compression rate and of the late ignition timing necessitated in order to avoid detonation. By removing the ignition plug to the centre of the cylinder, the same engine would run without detonation on any ordinary commercial petrol, while its best power output and efficiency were obtained with about 12° less ignition advance, indicating that the time taken to complete combustion was reduced to this extent or by about 0·0016 second. At the other end of the scale is an engine with overhead valves with no valve pockets and with the ignition plug fitted exactly in the centre of the cylinder head. This engine would run on any commercial petrol at a compression ratio of 5·4 : 1 and that without a trace of detonation, and in consequence gave a very high power output and efficiency. These, of course, are extreme instances, but they illustrate how important a part the position of the ignition plug actually plays.

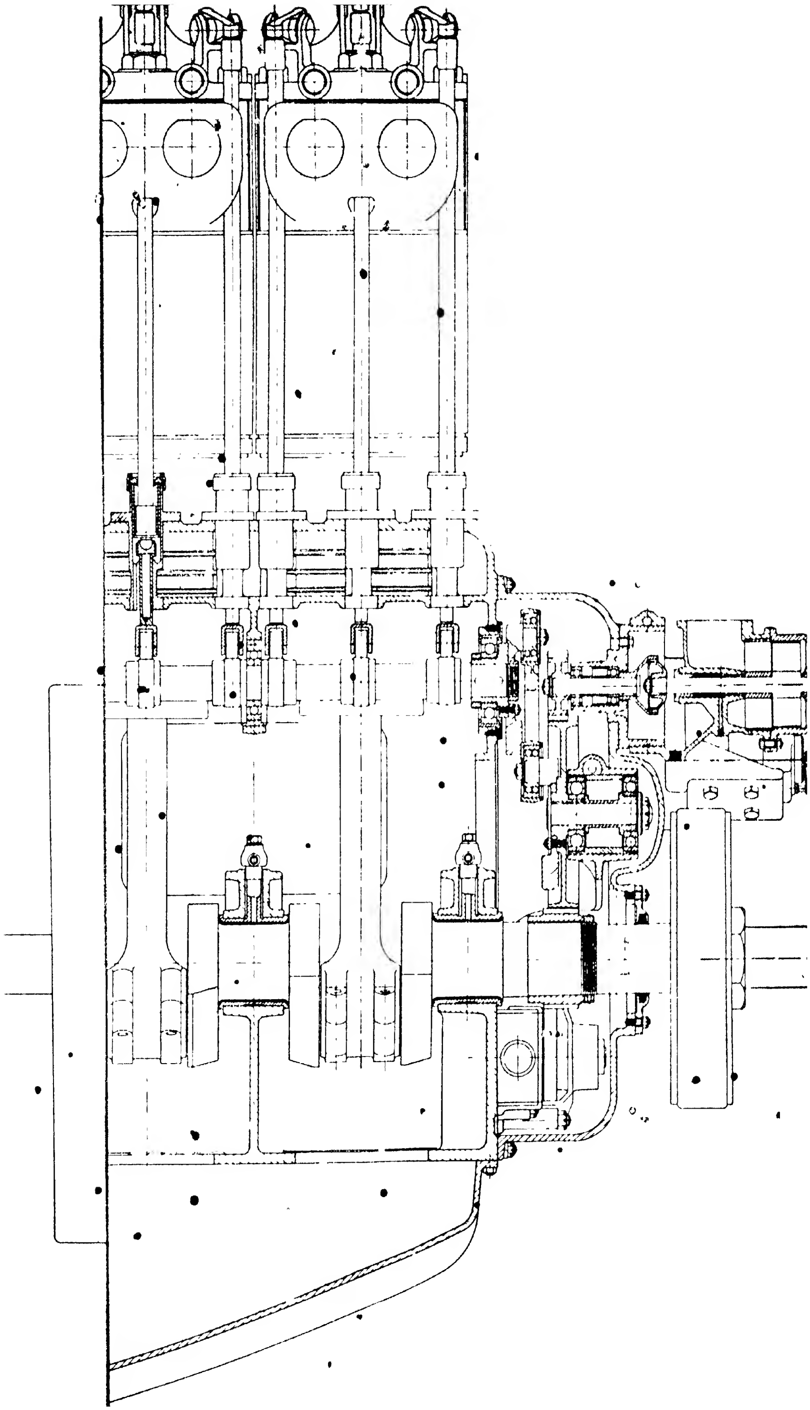
Apart from the question of detonation, the distance from the point of ignition to the farthest point in the combustion chamber controls, for an equal degree of turbulence, the time taken for inflammation to spread throughout the working fluid. This is very important, more particularly so on light loads, for, other things being equal, the rate of burning depends upon the flame temperature, and this, in turn, depends upon the proportion of residual exhaust products. As the load is reduced by throttling, so is the proportion of inert gases increased, and, in consequence, the flame temperature is reduced and with it the rate of burning. It may be argued that, within limits, the rate of burning matters but little, because it is always possible to advance the time of ignition to suit. This, however, does not quite meet the case, even on full load, for, if the period of combustion be prolonged unduly, the direct heat loss to the cylinder walls at this vital period may begin to assume serious

proportions, while when the load varies rapidly, as it does in all forms of motor traction, it becomes almost impossible so to adjust the time of ignition as to suit all conditions.

If the plug is placed centrally in the combustion chamber, a small ignition advance will serve for all conditions of speed or load, and the engine will run at all times at a high efficiency with fixed ignition timing. If, however, the plug is placed in such a position as to leave a long travel for the flame, not only will a considerable ignition advance be required at full load, but the engine will, at all times, be much more sensitive to the time of ignition, and will not run efficiently at light loads unless the ignition is advanced very much further than on full load. It should be noted that both the time taken to complete combustion and the tendency to detonate depend, for equal degrees of turbulence, upon the absolute, not the relative, distance from the sparking plug to the farthest point in the combustion chamber. Thus, both the time of combustion and the tendency to detonate will be nearly the same in a cylinder of 8-inch diameter with the plug in the centre as in one of 4-inch diameter with the plug at one side, or for similar plug positions both the tendency to detonate and the time of combustion will be greater with the larger cylinder. This factor would militate against the large cylinder were it not for the fact that in the larger cylinder turbulence is the better maintained, the gases are in more rapid motion, and combustion therefore spreads relatively faster. Hence the efficiency does not suffer, though the tendency to detonate necessarily remains greater in the larger cylinder.

That the valves shall open freely into the combustion chamber is so obvious as to need no comment, yet it is only too common to find that, in an endeavour to fit the largest possible valves, designers have apparently overlooked this point and have neutralized their efforts by failing to provide sufficient space between the valve head and the side walls of the chamber. To restrict the area of entry between the head of the inlet valve and the side walls of the combustion chamber is worse than to provide inadequate valve area, since it tends both to heat the incoming gases to a greater extent, and causes wire-drawing without the compensating advantage of additional turbulence, for the initial velocity of the entering gases is the more quickly damped out when they impinge against the rough walls of the combustion chamber.

In the design of the combustion chamber the three points to be aimed at are :



(1) To provide a chamber into which the valves open directly, so that after entry the gases need turn no corners and so lose their initial velocity;

(2) To arrange that the sparking plug be placed centrally in the combustion chamber;

(3) That there should be no valve or other pockets where the gases may become stagnant.

At first sight it might appear that the ideal combustion chamber would be a true sphere, but although this would be excellent from the point of view both of surface-volume ratio and of turbulence, it would be ideal only if the point of ignition were at the centre, which is obviously impossible. Since the point of ignition must of necessity be situated somewhere on or very near the surface,

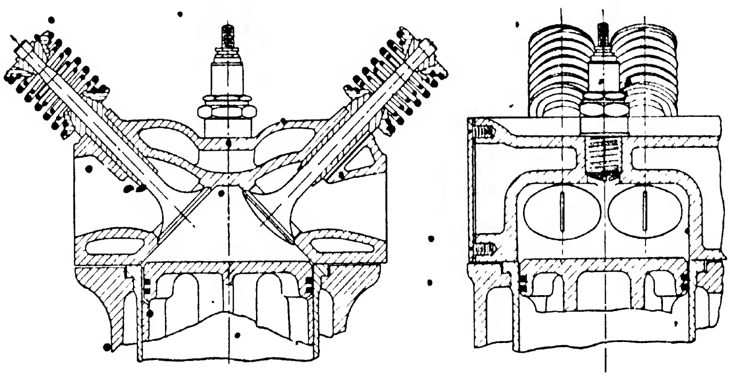


Fig. 24

it follows that the distance which the flame will have to travel will be equal to the full diameter of the sphere, and this will give rise to detonation and necessitate the use of a lower compression ratio, with consequent loss of power and efficiency. Probably the nearest practical approaches to the ideal combustion chamber are those shown in figs. 24 and 25. Fig. 24 represents the combustion chamber of a small racing engine which, though only $3\frac{1}{8}''$ bore and $3\frac{3}{4}''$ stroke, developed over 20 B.H.P. at a speed of 4300 R.P.M., while fig. 25 (see plate) shows a section of the cylinder of a large experimental engine $8\frac{1}{4}'' \times 9\frac{1}{2}''$ developing about 100 B.H.P. per cylinder at 1200 R.P.M., and an indicated thermal efficiency of 35·5 per cent on ordinary aviation petrol.

These forms, while almost ideal on thermo-dynamic grounds, are not very convenient from a mechanical point of view, because

they necessitate the use of overhead valves operated from two camshafts, for the central position of the sparking plug precludes the use of a central overhead camshaft. They are, however, particularly applicable to sleeve-valve engines, and the fact that they are inherent to this type of engine is, in the author's opinion, one of the strongest arguments in favour of the sleeve valve. For engines required to develop a very high power output and efficiency, similar forms of combustion chamber, but with the sparking plug placed at the side, are very commonly employed, and are, indeed, practically universal for all aircraft engines. The effect, however, of placing the plug at the side is to increase both the degree of ignition advance and the tendency to detonate, and so necessitate the use of a somewhat lower compression ratio and a consequent

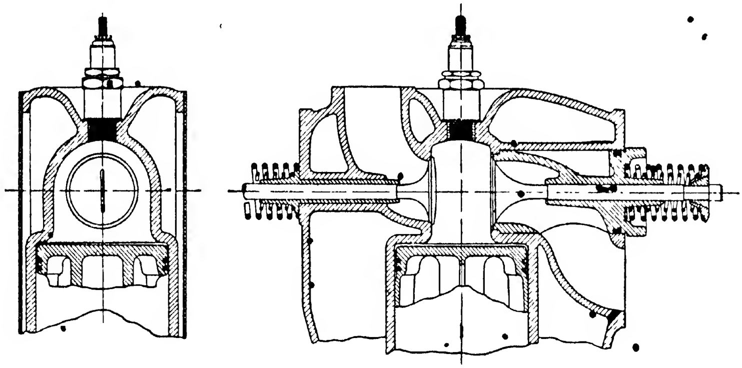


Fig. 26

reduction both in power and efficiency. By using two sparking plugs, however, placed at opposite sides of the combustion chamber, this disadvantage can largely be overcome, for the distance which the flame has to travel from the point of ignition until combustion is completed is then almost halved; but this assumes that the two plugs are synchronized perfectly, by no means a safe assumption when two magnetos are used, but probably quite safe when the primary circuit of two high-tension coils is broken by a single contact breaker.

The form of combustion shown in fig. 26, in which two horizontal valves are placed in a small cylindrical combustion chamber, is one of the best possible from the point of view both of detonation and combustion efficiency so long as the clearance between the piston and the cylinder head is reduced to the very limit necessitated by mechanical considerations, and so long also as the engine runs at a

comparatively low piston speed and therefore does not require large valves. It is therefore a particularly efficient form for short-stroke engines, but though excellent for power output and efficiency it is unfortunately very inconvenient from a mechanical point of view when applied to a vertical engine on account of the valve operation and pipe work.

The form shown in fig. 27, in which the inlet valve is placed vertically over the exhaust in a side-pocket, and in which both the piston and cylinder crown are concave, is surprisingly efficient in

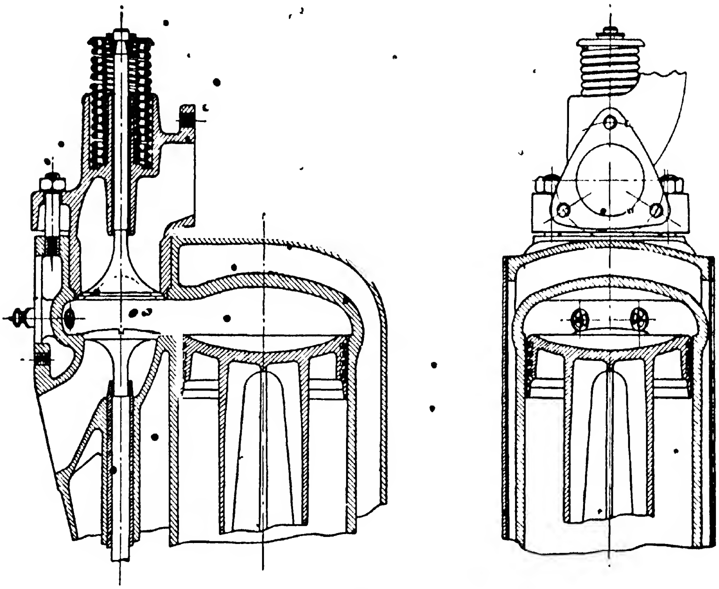


Fig. 27

maintaining turbulence despite the fact that the valves are pocketed, but, as may be expected with so long a travel for the flame, it is very bad from the point of view of detonation and therefore can only be used with a low compression ratio. With this form of combustion chamber and with the sparking plug fitted in the valve pocket, the tendency to detonate is considerable, while with the plug fitted on the side, remote from the valve pocket, it is one of the very worst possible forms, from the point of view of detonation, as indeed might be expected. This form, therefore, necessitates the use of a low compression ratio, but gives a very good power output and efficiency at that compression.

The form shown in fig. 28 is one which the author patented some years ago for use either with side by side valves or when one valve is fitted vertically over the other in a side pocket. In this type the whole of the combustion space is concentrated in the valve pocket, and there is a restricted communication with the cylinder, while the clearance between the piston and cylinder head is reduced to the lowest possible limit. The objects of this design are :

(1) To produce the maximum of turbulence by creating additional mechanical disturbance during the compression stroke, and more particularly during the last portion of this stroke when the gases entrapped between the flat piston and cylinder head are ejected violently into the combustion space.

(2) To combine the use of side valves with a compact and deep combustion chamber and one in which the sparking plug can be placed almost centrally while the extreme distance which the flame has to travel is reduced to the minimum.

This form of combustion chamber has proved almost comparable both in power output and efficiency with the overhead valve type, while, from the point of view of detonation, it is certainly very good. That in power output and efficiency it is not quite equal to the overhead valve type is due to :

(1) The presence of a thin layer of gas (from 4 to 8 per cent of the total quantity) entrapped between the piston and cylinder head. In this position it is so chilled that it does not burn until the piston has descended some distance; hence it is burnt at a lower efficiency.

(2) Since the valves do not open directly into the cylinder the volumetric efficiency is lower, though, since turbulence is created during compression, it is unnecessary to rely on the velocity through the inlet valves, and larger valves can be used when the cylinder centres will permit of this, or when the valves are placed vertically over one another instead of side by side, as is more usual.

Since the thickness of the layer of gas entrapped between the piston and cylinder crown is governed by purely mechanical considerations, and may be taken as constant, it follows that the proportion which this bears to the whole depends upon the length of stroke—the longer the stroke the more efficient does this form of combustion space become.

The form shown in fig. 29 may be taken as the conventional type for side-valve engines. It is bad, both from the point of view of turbulence and of detonation, with the result that not only is the efficiency and power output low for any given compression ratio,

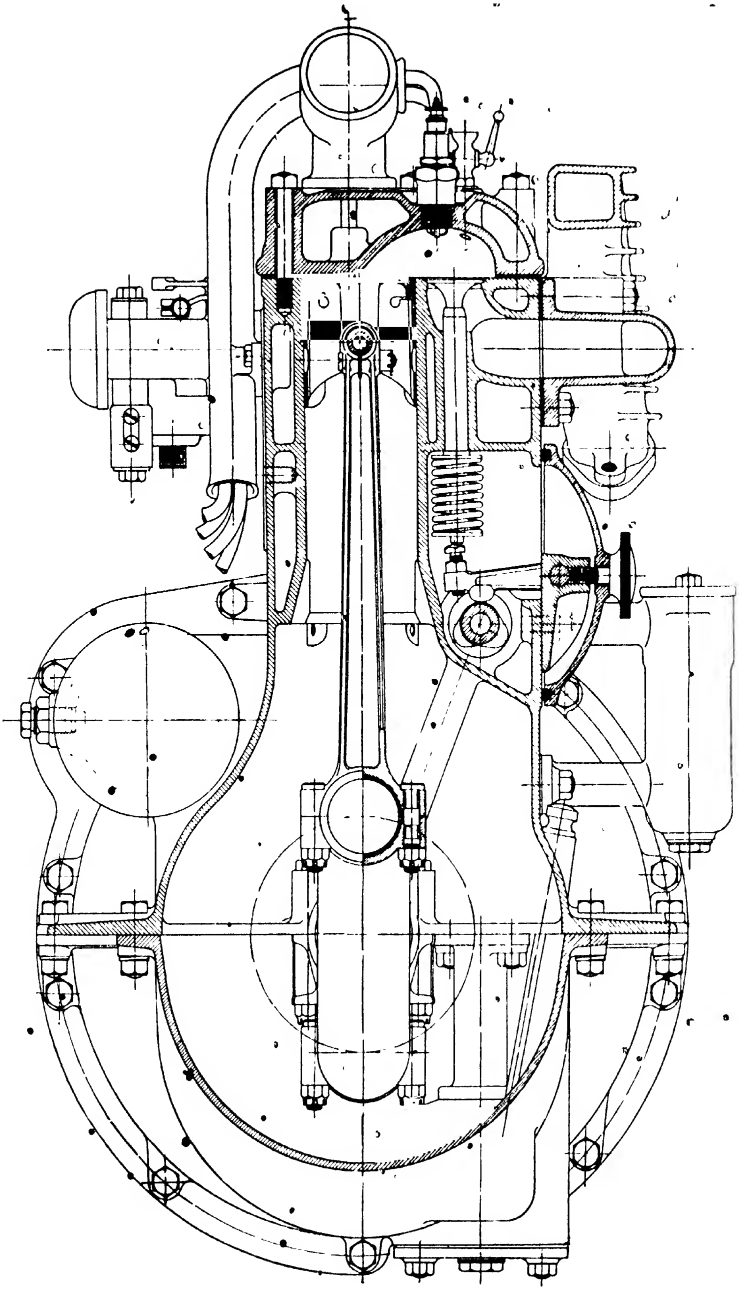


Fig. 28

but only a comparatively low compression ratio can be used, on account of detonation. To minimize the latter and generally to make the best of a poor form, the sparking plug should be placed as nearly in the centre of the combustion space as possible. This form of combustion space is particularly unsuitable for a short-stroke engine, because it then becomes very shallow, with the result that turbulence is still further reduced by surface friction, and, as the area of surface is very large, the stagnant layer of gas adhering to this surface and only partially burnt, or burnt late, and, therefore, very inefficiently, assumes a serious proportion of the whole, probably in many cases as high as 25 per cent.

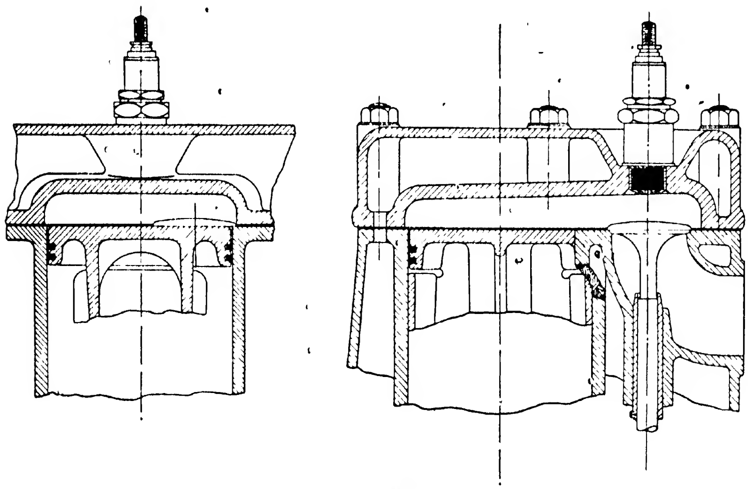


Fig. 29

The form shown in fig. 30 is also a conventional form, and from a mechanical point of view there is much in its favour for multiple-cylinder engines, since it permits at once of the use of large valves and close cylinder centres; it is convenient also from the point of view of pipe work and of valve operation; but when this has been admitted, there is little else in its favour, for, on thermo-dynamic grounds, it would be hard to find a worse shape. It affords the longest travel for the flame of any type, and it provides the maximum of surface and of surface friction, hence it is bad from the point of view both of detonation and turbulence, though, if the sparking plug be placed in the centre of the cylinder, it is but little worse from the former standpoint than the side-valve type.

The various forms of combustion chamber described above cover

practically all the possible shapes, with the exception of a few freak forms which have nothing to recommend them, and owe their origin merely to the desire to be unconventional.

It is difficult to assess the merits of the different types described above in quantitative terms, because so much depends upon the detail design of each particular example. The author has, at his laboratory, tested a large number of engines with each of the forms described above: out of these it is possible, by careful analysis, to select examples which are nearly similar in most respects other than the type of combustion chamber. The table on following page has been compiled from such tests of engines in which the gas velocity through the valves, the valve opening diagram, and several other conditions

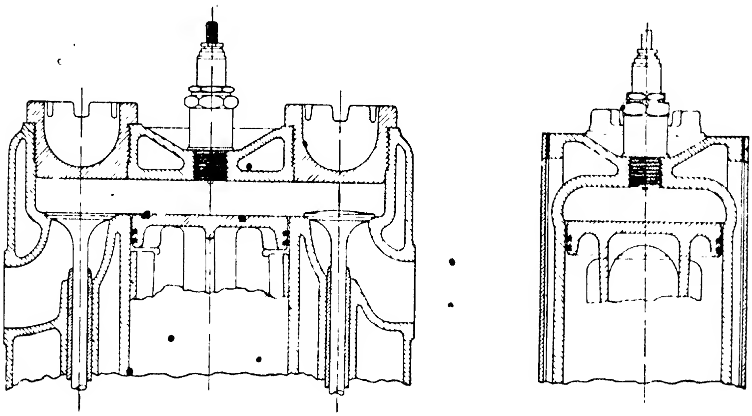


Fig. 30

were identical, while the mechanical losses in all cases were determined, and the highest compression ratio consistent with freedom from detonation on a standard petrol was determined also. In this table it is assumed that with each type of combustion chamber the compression ratio was such that detonation could just be avoided: this was found in each instance by using a special petrol which detonated with extreme readiness, and by adding just sufficient toluene to check detonation, the relation between the proportion of toluene needed and the highest safe compression ratio having been determined previously by tests on the author's variable compression engine.

The figures given in this table are corrected also for cylinder capacity in accordance with the results quoted in Chapter III. This correction is necessarily somewhat empirical, but since, in all

Type of Combustion.	Comp. Ratio	Relative Power Output and Efficiency.
		Per cent.
Four valves pent roof with sparking plug in centre	5·4 : 1	100
Five valves vertically in head, sparking plug in centre, stroke-bore ratio 2 : 1	5·4 : 1	100
Valves vertically in head, 2 sparking plugs at opposite sides	5·2 : 1	97
Valves vertically in head, 1 sparking plug at side ...	5·0 : 1	94
Inlet over exhaust, sparking plug in centre... ...	4·7 : 1	87
Inlet over exhaust, sparking plug in valve pocket...	4·5 : 1	85
Inlet over exhaust, sparking plug opposite valve pocket	4·2 : 1	80
Side valves, special high turbulence design, fig. 28, stroke-bore ratio 1·4 : 1	4·9 . 1	88
Side valves, special high turbulence design, fig. 28, extra long stroke, ratio 2/1 but otherwise similar	5·0 . 1	90
Side valve conventional type, with plug in centre of combustion space	4·6 : 1	80
Side valve conventional type, with plug directly over inlet valve	4·4 : 1	77
Tee head, fig. 30, with plug in centre of cylinder head	4·6 : 1	80
Tee head, with plug directly over inlet valve ...	4·2 . 1	75

cases, the cylinder size did not vary greatly, the influence of the correction is, in any case, a very small one.

For convenience, the best form of combustion chamber is taken as 100 per cent efficient and the relative power output and efficiency of the others is expressed in terms of percentage of the best example.

The comparative figures given in the above table are, of course, purely empirical, and are based on average results. It must not be supposed that of any one general type of combustion chamber a more efficient example cannot be made by dint of careful design; but this, of course, applies to all types, and does not affect the comparison except in so far that there is naturally a somewhat greater scope for improvement in the less efficient types.

As explained previously, the limiting compression ratio is dependent to some extent upon the size of cylinder, because cylinder size is one of the factors which determines the distance the flame has to travel from the sparking-plug points. The figures in the above table are, however, all based on the assumption that the cylinder

capacity is in the neighbourhood of 100 cub. in. in each case, and 4 to 5 in. bore.

Determination of the Combustion Chamber Efficiency.—It has been shown that the efficiency of the combustion chamber is the most important of all the factors which control both the power output and the thermal efficiency of an engine. Unfortunately, however, it is, in most cases, very difficult to determine even the relative efficiency of two different designs of combustion chamber, particularly in the case of multiple-cylinder engines, because irregularities in distribution play so important a part as to render a determination based on the known consumption of fuel per horse-power hour of very little value. It is very commonly found that of two engines the measured fuel consumption per horse-power hour is the same, despite the fact that one may have a considerably more efficient combustion chamber than the other. At first, it would appear that the design of both engines is equally efficient, whereas in fact the former is much the more efficient of the two, but the practical value of its high inherent efficiency has been annulled by greater irregularities in distribution. Were it practicable to change the whole distribution system from one engine to the other a very marked change in efficiency would at once be observed; but this in practice is seldom, if ever, possible, because in almost all existing designs of multi-cylinder engines a considerable proportion of the induction passage is embodied in the cylinder casting, so that the whole distribution system is seldom, if ever, interchangeable as between two engines with different forms of combustion chamber, hence it is often very difficult to determine the relative merits of two different forms, large though the difference may be.

The most practical method which the author has yet found for determining the true efficiency of a multi-cylindered engine independently of all irregularities in distribution, defective carburation, etc., depends upon the fact that in the case of any hydro-carbon fuel or alcohol the heat energy liberated by the combustion of any given weight of air is almost exactly the same, however much excess of fuel may be present, provided always that there is an excess of fuel, or, in other words, so long as the mixture is over-rich the thermal efficiency, based on that portion which is burnt, will be the same over a very wide range of mixture strength on the rich side, with the result that over the range from 5 per cent rich to 35 per cent rich the heat liberated by unit weight of air is constant to within extraordinarily narrow limits

If, now, instead of measuring the weight of fuel consumed, we measure the air consumption and calculate the efficiency obtained in terms of lb. of air consumed per horse-power hour, we shall arrive at a means of determining the true thermal efficiency of an internal-combustion engine, irrespective of any losses due to irregularities in distribution, defective carburation, precipitation of liquid fuel, etc. If, by air measurement, we find that the thermal efficiency of an engine is low, then we know that the combustion chamber design is at fault, and that no amount of juggling with the carburettor or distribution system will avail.

On the other hand, as so often happens, an engine apparently well designed and with what would appear to be an efficient form of combustion chamber shows on test a very poor fuel economy; in such a case, we can, by ascertaining the air consumption, determine at once whether the fault lies in the engine design or in that of its distribution system. Given reliable means for measuring accurately the air consumption, it is necessary only to set the carburettor to give a rich mixture, in practice to set it to the mixture giving maximum power (which on petrol is about 15 to 20 per cent rich) and take readings of air consumption ; all that is needed being to ensure that the mixture strength is such that the weakest cylinder in the group is receiving a mixture not less than about 5 per cent rich. Since there is an extremely wide range available beyond this, i.e. up to about 40 per cent rich (after which combustion becomes delayed again as on the weak side), this presents no difficulty whatever.

The absolute value found for the thermal efficiency by this method is, in all cases, somewhat higher than that obtained from the fuel-consumption readings when working with the most economical mixture strength, and this is so even in the case of single-cylinder engines, where distribution losses can be almost entirely eliminated, the difference being due to :

(1) The larger increase in specific volume when an over-rich mixture is used.

(2) The slight loss due to condensation of liquid fuel on the cylinder walls which escapes combustion and ultimately finds its way past the piston into the crankcase.

(3) The small loss due to precipitation of liquid fuel on the walls of the induction pipe; this fuel enters the cylinder in gulps of liquid which are not completely vaporized, at any period during the cycle.

(4) The small loss of liquid or vapour due to "blow-back," or

reverse flow in the induction pipe due to the sudden closing of the inlet valve.

Once the air consumption is known, the thermal efficiency of any engine can be determined from the formula

$$E = \left(\frac{\text{I.H.P.}}{\text{lb. of air per hour}}\right) \times C,$$

where C is a constant representing the amount of heat liberated by the combination of 1 lb. of air; for all petrols it may be taken as 196.

The following table gives the effective calorific value, the mixture strength for complete combustion in terms of air/fuel ratio by weight, the amount of heat liberated by the combination of 1 lb. of air, and the value of C in the formula above, for a representative selection of fuels, from which it will be seen that over the whole range of available fuels the heat liberated by the combination of 1 lb. of air is substantially the same.

The figures given in the fourth column are calculated as for a

TABLE I

Fuel.	Effective Lower Calorific Value B.Th.U s per lb.	Air Fuel Ratio.	Heat liberated by 1 lb. of Air B.Th.U.s.	Value of Constant.
Petrol samples—				
(1)	19,200	15·05	1275	197·0
(2)	19,020	14·7	1295	195·0
(3)	19,120	14·8	1293	195·5
(4)	18,900	14·6	1295	195·0
(5)	19,090	14·9	1282	197·0
(6)	19,250	15·0	1285	196·5
(7)	18,920	14·7	1288	196·0
Kerosene	19,100	15·0	1275	197·0
Hexane	19,390	15·2	1275	197·0
Heptane	19,420	15·1	1285	196·0
Benzene	17,430	13·2	1320	192·5
Toluene	17,660	13·4	1315	193·0
Cyclohexane ...	18,940	14·7	1290	196·0
Heptylene... ...	19,320	14·7	1320	192·5
Ether	16,830	13·0	1295	195·0
Ethyl Alcohol—				
99 per cent ...	11,950	8·95	1333	190·0
95 " ...	11,125	8·4	1330	190·5

mixture giving complete combustion, as also the value of the constant C; but over a very wide range of mixture on the rich side, the variation is, in all cases, very small indeed, and appears to be almost exactly the same in the case of all volatile liquid fuels, so that the possible error due to variations in mixture strength is extremely small.

The following tables, Nos. II to IV, give the results of typical

TABLE II

Petrol R = 5 : 1, Sample No. 4

Date of test, 5/7/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct	196·0	132·0	32·0	6·13	32·1
Plus 5 % excess fuel	196·5	135·0	32·7	6·0	32·8
,, 10 ,,	197·0	136·5	33·1	5·95	33·1
,, 15 ,,	197·5	137·5	33·3	5·93	33·2
,, 20 ,,	198·1	138·0	33·4	5·94	33·15
,, 25 ,,	198·8	138·0	33·4	5·96	33·05
,, 30 ,,	199·5	137·5	33·3	5·9[illegible]	32·85
,, 35 ,,	200·0	136·5	33·1	6·04	32·7

Maximum thermal efficiency calculated from fuel consumption 32·1 per cent, with mixture 16 per cent weak.

TABLE III

Ethyl Alcohol 99 per cent R = 5 : 1

Date of test, 27/8/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct	199·0	141·0	34·2	5·82	32·9
Plus 5 % excess fuel	199·5	143·0	34·6	5·77	33·3
,, 10 ,,	200·0	144·5	35·0	5·72	33·65
,, 15 ,,	201·0	145·5	35·25	5·70	33·7
,, 20 ,,	202·0	146·5	35·5	5·69	33·75
,, 25 ,,	203·0	147·0	35·6	5·70	33·7
,, 30 ,,	204·0	147·3	35·7	5·72	33·65
,, 35 ,,	205·0	147·6	35·8	5·73	33·6

Maximum thermal efficiency calculated from fuel consumption 33·0 per cent, with mixture 15 per cent weak.

TABLE IV

Ethyl Alcohol 95 per cent R = 5 : 1

Date of test, 16/8/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb per sq. in.	Indicated Horse-Power.	Lb of Air per I H.P. Hour.	Indicated Thermal Efficiency.
					Per cent
Correct	203·0	145·0	35·1	5·79	33·1
Plus 5 % excess fuel	203·5	146·7	35·5	5·72	33·6
,, 10 ,,	204·0	148·0	35·8	5·71	33·65
,, 15 ,,	205·0	149·0	36·0	5·70	33·7
,, 20 ,,	206·0	149·5	36·15	5·70	33·7
,, 25 ,,	207·0	150·0	36·25	5·70	33·7
,, 30 ,,	208·0	150·3	36·3	5·71	33·65
,, 35 ,,	209·0	150·5	36·35	5·71	33·1

Maximum thermal efficiency calculated from fuel consumption 32·9 per cent, with mixture 15 per cent weak.

tests on petrol and alcohol (the latter both nearly pure and 95 per cent) over a wide range of mixture strength from correct to 35 per cent over-rich, taken on the variable compression engine described in Chapter II, while at the foot of each table the maximum thermal efficiency as deduced from the fuel consumption at the most economical mixture strength is given for purposes of comparison.

In Table V are given the results of air measurement tests at varying compression ratio, ranging from 4·0 : 1 to 7·0 : 1, the fuel used in this case being benzol. Here again the agreement between the calculated thermal efficiency from the air consumption with a rich mixture, and from the fuel consumption with a weak mixture, is on the whole very consistent.

Similar tests carried out on a six-cylinder aero-engine with a compression ratio of 4·7 : 1 showed an air consumption of 6·4 lb. of air per I.H.P. hour over a range of mixture strength from 10 per cent to 25 per cent over-rich, using standard aviation petrol. The air efficiency in this case works out at 31 per cent, a figure very considerably greater than that obtained from the fuel consumption, which was only 28·2 per cent at the most economical mixture strength. The discrepancy in this case represents the loss due to irregularities in distribution, etc. A comparison of these results with those given previously for the variable compression research engine is very interesting, all the more so because both engines have

TABLE V

Varying Compression Ratio

Fuel, Benzol, about 20 per cent rich

Date of test, 16/10/21

Compression Ratio.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency as found by Air Measurement.	Indicated Thermal Efficiency as found by Fuel Measurement 15 per cent weak.
					Per cent.	Per cent.
4	203	125·0	30·3	6·70	28·8	27·7
5	195	136·5	33·1	5·90	32·8	32·0
6	189	145·0	35·2	5·37	35·9	35·0
7	185	152·0	36·8	5·03	38·3	37·3

nearly identical forms of combustion chamber, very nearly the same cylinder capacity, and, in both cases, the charge is ignited by sparking plugs at opposite sides of the combustion chamber. That the aero-engine does not show, by the air-consumption test, so high an efficiency as the research engine at the same compression ratio, namely, only 31 per cent as against 31·7 per cent, is to be explained probably by :

(1) Although two plugs were used they were sparked by two separate magnetos, and therefore not so accurately synchronized.

(2) The aero-engine had a relatively shorter stroke, so that the combustion chamber was flatter, and therefore somewhat less efficient.

(3) The mechanical condition of the aero-engine was probably not so good, i.e. there was probably more leakage loss.

The difference in fuel consumption was, however, much more marked, for the maximum thermal efficiency was 30·9 per cent reckoned on the fuel consumption at a compression ratio of 4·7 : 1 in the case of the single-cylinder research engine and only 28·2 per cent in the six-cylinder aero-engine. After allowing for the different combustion-chamber efficiencies of the two engines, the loss by irregularities in distribution, loss of unburnt fuel, etc., of the aero-engine is about 10 per cent, while in the single-cylinder it is only about 2·5 per cent. The indicated mean pressure was found to be exactly the same in both engines, namely, 133 lb. per square inch ; but measurements of volumetric efficiency showed that, while the research

engine had a volumetric efficiency at this compression ratio of 76·2 per cent, that of the aero-engine was 78 per cent, a difference which very approximately compensates for the lower combustion chamber efficiency of the latter engine. Again, tests carried out on a four-cylinder commercial vehicle engine with a compression ratio of 4·24 : 1 showed an air consumption of 7·75 lb. per I.H.P. hour = 25·6 per cent thermal efficiency as against 6·58 = 30·4 per cent in the variable compression engine at the same ratio, from which it may be deduced that the efficiency of the combustion chamber was only 85 per cent that of the research engine at the same comparison ratio. In this particular engine the design of the cylinder head was very defective, the valves being placed in deep-set shallow valve pockets with the sparking plugs directly over the inlet valves. The fuel efficiency at the most economical mixture strength was, however, about 23·8 per cent, showing that the efficiency of distribution, if such a term may be used, was, in the case of this four-cylinder engine, as high as 93 per cent. In other words, this engine made up, to some extent, so far as fuel efficiency was concerned, for bad cylinder design by having a quite unusually efficient distribution system, but the bad cylinder design showed itself in the low power output obtainable, the indicated mean effective pressure being only 93 lb. per square inch as against 129 lb. per square inch in the variable compression engine under exactly similar conditions. Had the volumetric efficiency been the same in both cases, the mean effective pressure would have been $\frac{85}{100} \times 129$, or about 110 lb. per square inch; that it was, in fact, only 93 lb. was due again to defective cylinder design, whereby the free entry of the gases after leaving the inlet valve was obstructed by the surrounding walls of the shallow valve pockets. Air measurements showed also that, while the volumetric efficiency of the research engine at this compression ratio, and at the same temperature, was 77 per cent, that of the four-cylinder engine was only 66 per cent, so that the maximum M.E.P. should have been $\frac{66}{77} \times 110$, or 94 lb. per square inch, a figure which agrees very closely with the 93 lb. actually measured during the tests. This latter is rather a striking example of an inherently defective engine, which showed a comparatively good economy because its induction system was unusually efficient.

In the absence of any means for measuring the air consumption, a fair estimate of the efficiency of different forms of combustion

chamber can be gained by comparing the maximum mean effective pressure, but this again assumes that the volumetric efficiency is the same in both instances.

Such an assumption is, of course, not always justifiable, but it is at least fair to assume that the variations in volumetric efficiency as between two somewhat similar types of engine will be very much less, and will have a much smaller influence on the determination of the combustion-chamber efficiency, than the variations in mixture strength, as between individual cylinders, despite the rather exceptional example quoted above. Where means are available for air measurement, the efficiency of any form of combustion chamber can very readily be determined from the measured air consumption.

If an engine consumes its air efficiently, then it is an efficient engine, and to render it economical in fuel is a question solely of carburation and distribution. If its air consumption is heavy, then no amount of finessing with carburettor adjustments or distribution design will render it efficient.

CHAPTER V

LUBRICATION AND BEARING WEAR

Three factors have to be considered in the design of any bearing: first, the ability to carry the necessary duty in the space available with a reasonable margin of safety against breakdown; next, the rate of wear of the bearing surfaces; and, finally, the energy lost in friction.

Piston friction will be dealt with in a later chapter, so that the bearings only will be considered here.

These are generally of the "plain" or sliding friction type. Ball and roller bearings, being of a fundamentally different nature, are not included.

Where two surfaces, apparently in contact, are moving relatively one to the other, there are three possible cases to be considered.

In the first case, that of "dry" friction, the surfaces are in actual contact without any lubricant. In this case the friction is very great, and only very low loads and speeds can be imposed without seizure. This case never occurs when a bearing is functioning properly, so need not be dealt with further.

The second condition, that of "greasy" friction, occurs when the surfaces, though virtually in contact, are actually lubricated with some substance which discourages their mutual adhesion. The lubricant in such cases appears to function by exerting some kind of chemical action on the metallic surfaces.

Lastly, there is "viscous" friction, in which the surfaces are completely separated by a film of lubricant: this is clearly by far the most desirable state of affairs, and, fortunately, it is one which can easily be attained in a well-designed and adequately lubricated bearing, and may, in fact, be regarded as the normal condition.

In a journal bearing the necessary oil film between, the loaded surface is maintained by a wedging action, due to the fact that the shaft sets itself eccentrically in its journal. This is made clear in fig. 31, where it can be seen that the oil in the wide space A is

dragged by the rotation of the shaft into the narrow space B, thus forcing the surfaces apart. Under these circumstances the frictional loss and the thickness of the oil film for any given conditions can be theoretically evaluated, and it has been found that the results so obtained hold good in practice so long as the bearing is not too heavily loaded. These two factors—film thickness and friction—depend, for any given bearing, solely on the load, the speed, and the viscosity of the lubricant.

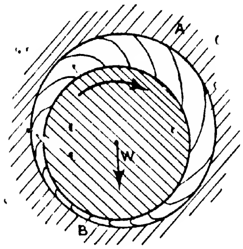


Fig. 31

The influence of these three factors is roughly as follows: Increase of load, *alteris aequis*, increases the friction, though not nearly in direct proportion, and decreases the film thickness. Increase of speed increases both the friction and the film thickness, as does also an increase in the viscosity of the lubricant. However, increase of either load or speed, by increasing the rate of energy loss, heats up the oil, and thus decreases its viscosity.

This fact, in the case of an increase of load, to some extent nullifies the increased friction; it still further decreases the film thickness, while with increased speed the theoretically increased film thickness is actually reversed. Thus both increased load and speed tend to reduce the film thickness.

It is clear that, as no surfaces are perfectly smooth, there is a limiting thickness of oil film at which the high spots of the two surfaces begin to bridge across the oil film.

At these points the oil film is no longer of appreciable dimensions, so that the laws of "greasy friction" begin to apply. The frictional force, under these circumstances, is, at such points, considerably higher than with viscous friction, and obeys totally different laws. The chief factor appears to be a property, probably of a chemical nature, called "oiliness," which tends to reduce friction between two surfaces in contact. It is quite distinct from viscosity, as quite thin oils, such as sperm or rape, can have it in a far greater degree than far more viscous substances, such as treacle, from which it appears to be entirely absent. Unfortunately the data on this subject is very limited and largely contradictory, but it is fairly well established that oils of animal or vegetable origin, such as sperm, rape, and castor oils, are considerably "oilier" than the hydrocarbon mineral oils, while viscous substances of a "sticky" nature are entirely devoid of the property. The nature of surface metals appears also

to exert an influence, but there is no agreement as to the relative virtues of any particular metals.

It is clear that if the lubricant has but little "oiliness," the friction at the "high spots" will be exceedingly high, and that the heat so generated will still further reduce the thickness of the oil film, thereby making matters worse, and so starting a "vicious circle" ending in seizure.

With an oily lubricant, on the other hand, the increase in friction at the "high spots" will be much less, and the risk of seizure correspondingly deferred.

The influence of "oiliness" is thus only of importance where, owing to excessive loading, unsuitably disposed surfaces, or inadequate oil supply, a complete film of oil cannot be maintained. It has apparently no influence on the friction under normal conditions, though it may enable one to use, in any given case, a thinner oil, with correspondingly reduced friction, by relying, as a reserve in case of abnormal conditions, upon oiliness rather than upon excess viscosity.

To return to the consideration of normal conditions, the maintenance of an adequate oil film necessitates efficient arrangements both for the removal of the heat generated and for the continuous replenishment of the oil in the bearing.

The heat is got rid of, to some extent, by the oil which forms the actual oil film, but mostly either by conduction from the bearing surfaces, or by excess oil which runs through or over the bearing without actually forming the load-carrying film. The advantages of forced lubrication are mainly due to the cooling effect of the excess lubricant.

The supply of oil to the bearing is greatly facilitated by the "pumping" action caused by alternating loads, it being found that any given bearing will carry an alternating load considerably in excess of the maximum steady unidirectional load. Again, a narrow bearing loses a far greater proportion of its oil through side leakage than does a wide one, so is correspondingly less efficient when subject to "natural" lubrication, while with forced lubrication a narrow bearing will carry a heavier continuous load per square inch of projected area because of the greater quantity of oil which can be circulated through it and therefore the better cooling. It is also necessary that the surfaces should be of such a shape that the wedging action, which maintains the oil-film at the highly loaded parts of the bearing, can take place, for the pressure which must exist in order

to carry the load is far in excess of any that a forced lubrication system can supply.

Fortunately, journal bearings are naturally suitable, as explained earlier, but with thrust collars special measures have to be taken, as in the Michel bearing. However, in practically all cases, in so far as high-speed internal-combustion engines are concerned, such thrust loads as are involved are most conveniently dealt with by the adoption of ball thrust faces.

When conditions are too severe for even a molecular oil film to survive, the surfaces come into actual metallic contact with one another that is to say, when the oil film becomes so thin that the two surfaces come within the zone of molecular attraction, an exceedingly high temperature is at once set up, resulting in local fusion of the surfaces. In some cases such local fusion may, by removing a high place in the surface of the bearing, relieve the loading at this particular point and so permit of the re-formation of the oil film. Unless this occurs, and the oil film re-forms immediately, the surface fusion will spread until the bearing either seizes solid or the bearing metal melts and runs out. When both bearing surfaces are hard and of anything approaching the same melting point, the surface fusion is generally accompanied by partial welding, and the two surfaces become inextricably locked. When the material forming one surface is relatively soft and has a low melting point, as in the case of white metal, the bearing merely melts, either locally or completely; in the former case, as already pointed out, local melting may be due to the existence of a high spot, and the fusion of this high spot may at once relieve the cause of trouble and permit of the restoration of the oil film in time to prevent any further spread of surface fusion: this is possible because the melting point of white metal is well below the boiling point of the lubricant. Local fusion is a very common occurrence in white metal lined bearings, especially when new and not too well fitted, and is often quite harmless.

The intensity of heat flow when an oil film breaks down locally is very surprising; it is quite common to find two case-hardened steel surfaces fused together locally and the temper of the surfaces undisturbed within less than ½ in. from the point of fusion. This corresponds to a temperature difference of something like 2500° F. in ½ in.

The case of bearings submitted to very heavy loads and low rubbing speeds is somewhat different. Here the time element

enters prominently into the question. Since at low speeds the wedging action, tending to maintain the oil film, is very slow, the effect of the pumping action, due to change of load, is of correspondingly increased importance. There is abundant evidence that under severe loads and favourable conditions even bronze bearings may be crushed without breaking down the oil film. So long as facilities are available for the replenishment of the oil, and so long as the rubbing velocity is such that the product of loading and rubbing velocity does not exceed a certain figure that is to say, so long as the heat generated does not exceed the rate at which it can be dissipated by replenishment with cool oil or by conduction—a journal bearing will not fail from pressure. Failures of heavily loaded, slow-moving journal bearings are almost invariably due either to an interruption in the oil supply or more frequently to bending or distortion of one of the members, causing excessive local pressure and heat flow. The only bearings in a high-speed internal-combustion engine submitted to very heavy loading with low rubbing velocity are the gudgeon-pin bearings. Experience has shown that when the gudgeon-pin is supported in such a manner that it does not distort appreciably through bending, maximum pressures up to 6000 lb. per square inch may quite safely be carried, without forced lubrication and without perceptible wear.

Wear of Bearings and Shafts. In view of the fact that most bearings are completely oil-borne, it is perhaps a little surprising at first sight that wear should take place at all, since the two surfaces never actually come into contact. The explanation appears to be that all wear is due to the abrasive action of small particles of grit carried by the oil. These particles, which are, for the most part, so small that they cannot be removed by filtration, are carried by the oil into the bearing and there embed themselves in the softer of the two surfaces; thus in a white metal lined bearing the particles of grit invariably embed themselves in the soft white metal. Once partially embedded they proceed to lap the shaft. It is clear that only those particles of grit which project far enough from the softer material to span the oil film and so actually touch the other member can cause wear. Other things being equal, therefore, the rate of wear depends upon the thickness of the oil film, which, in turn, depends upon the pressure and temperature the cooler the lubricant, or the lighter the pressure, the greater is the thickness of the oil film and, therefore, the greater the distance across which the particles of grit must reach before they can come in contact with the harder member.

Again, the rate of wear depends upon the hardness of the surfaces of the materials. It is common knowledge that when a soft-steel shaft runs in white-metal bearings it is the shaft and not the bearing material which wears; this is perfectly natural, since the particles of grit will always tend to embed themselves in the softer surface of the two and so proceed to cut or lap the other and harder surface. In order to reduce wear it is desirable that the difference in hardness between the two surfaces should be as large as possible: thus in the case of a steel shaft running in white-metal bearings, the softer the white metal the more readily will the particles of grit embed themselves entirely out of harm's way, while the harder the shaft the less readily will it submit to the cutting action of such particles of grit as are not completely embedded.

All available evidence indicates that the once popular idea that a hard white metal should be used, and that its surface should be rendered even harder by hammering or driving a taper mandrel through it, is quite erroneous—the surface of the white metal should be as soft as is consistent with the necessary resistance to crushing. In the case of two very hard surfaces, such as case-hardened steel against cast-iron or hardened steel, very little wear occurs, the probable explanation being that the hardness of both surfaces exceeds that of the particles of grit, so that the latter are merely ground up between the two surfaces and do not get any opportunity of embedding themselves in either or of cutting them.

It has been observed that such bearing surfaces, even when freely exposed to road dust, etc., do not wear readily, but that when carborundum is introduced along with the lubricant very rapid wear takes place.

It is when two surfaces, both relatively soft and of somewhat similar hardness, are employed that the most rapid wear takes place. A soft-steel shaft running in bronze bearings wears very rapidly indeed, unless the load factor is so low as to permit of a very thick oil film being maintained.

One very striking example of excessive wear between two surfaces of nearly similar hardness is to be found in the case of copper aluminium alloys and phosphor-bronze. An alloy consisting of 88 per cent aluminium and 12 per cent copper affords an excellent bearing material for hardened-steel shafts. It is light, which is often important, is an excellent conductor of heat, and is readily cast and machined. When, however, a phosphor-bronze shaft is run in bearings of this aluminium alloy, the shaft, which is slightly

the harder of the two, wears away with almost incredible rapidity. Similarly, experiments which the author has carried out with bronze piston rings in an aluminium cylinder and with an aluminium piston in a bronze-lined cylinder resulted, in the former case in the rings wearing down to half their original thickness in eight hours, and in the case of the bronze-lined cylinder the liner wore about 0·010 in. oval in a run of twelve hours. In neither case did the aluminium show any appreciable wear.

Again, a soft-steel shaft, running in a copper-aluminium bearing, wears away very rapidly.

Probably the worst possible results are to be found when two similar and relatively soft materials are used for the two members of a bearing, for then not only is the difference in surface hardness reduced to zero, but the opportunities for welding together in the event of a failure of the oil film and consequent local fusion are at a maximum.

In the case of cast-iron or hardened steel, both surfaces are so hard as to be very little affected by grit, but, in the event of a breakdown of the oil film, the two are very liable to become welded.

Experience with aluminium pistons has shown :

(1) That when these are fitted in soft-steel cylinders the cylinder bore wears very rapidly ;

(2) When fitted in hard-steel cylinders—0·4 carbon—the wear is very slight ;

(3) When fitted in cast-iron cylinders finished by grinding, wear of the cylinder bore takes place if the grinding material has not been thoroughly removed. Such wear does not take place when the cylinders are reamed or when lapped after grinding.

Maximum Pressures on Bearings.—So far as high-speed bearings are concerned—that is to say, when the rubbing velocity exceeds about 8 ft. per second—the load factor only, that is, the product of load and speed, need be taken into account; the maximum pressure, so long as it is not high enough to distort or crush the bearing material, is of little moment, since it is not applied for a long enough period to have any influence on the conditions of lubrication.

Limiting Load Factor.—The highest load factor which can safely be carried by a bearing depends upon :

(1) The system of lubrication, whether forced, natural, or fed with a measured quantity of fresh oil.

(2) Both the viscosity and the oiliness of the oil.

(3) The facilities available for conducting away the heat generated in the bearing.

Under the most favourable circumstances, with forced lubrication, and good facilities for dispersing the heat, load factors as high as 20,000 lb. ft. per second can be safely carried in a journal bearing with alternating load. With such a load factor the rate of wear is of course considerable, but there are many examples of crankshaft centre bearings in aero-engines with load factors as high as this.

Where very high rubbing velocities are involved, much higher load factors can be carried when floating bushes are employed. Such bushes floating freely between the two members, rotate at an intermediate speed, so that the rubbing velocity between either face is halved; also they permit of a much greater circulation of cooling oil through them. Under such conditions the load factor may be increased by 50 per cent without imperilling the bearing.

Maximum Load.—Where the rubbing velocity is low the only limit to the maximum load is set by the rigidity of the members. There is no danger of the oil film being broken down by pressure alone, provided there is scope for natural replenishment. When serious distortion takes place, the load factor may be increased locally—that is to say, the pressure may all be concentrated on one point in the bearing; and since the rubbing velocity is the same at all points, it follows that the product of pressure and rubbing speed may be excessive at one point, causing rapid local heat flow and ultimate breakdown of the oil film.

Proportions of Bearings.—From the point of view of reducing the load factor it is clear that the requisite area of bearing surface should be obtained by lengthening the journal rather than by increasing its diameter; since any increase in diameter involves a corresponding increase in rubbing speed. On the other hand, from the point of view of rigidity and freedom from distortion a long journal is very undesirable, since the load cannot be evenly distributed over it, nor can so much cooling oil be circulated through it owing to the greater resistance imposed. With the order of loads that obtain in the bearings of a high-speed internal-combustion engine, the pressures are so high that it is seldom practicable on the score of distortion to make any journal wider than one and a half times its diameter. With "natural" as opposed to forced lubrication very narrow journals are objectionable, because the area for the escape of oil so greatly exceeds the area available for replenishment.

Generally speaking it is desirable so to arrange the main bearings that the width of the journal is about equal to the diameter for forced lubrication and about one and a half diameters for natural or splash lubrication.

Load Factor and Wear.—Other things being equal, the rate of wear may be taken as being almost directly proportional to the load factor. It depends very largely on the facilities for cooling; thus, in the case of crankshaft bearings, for equal load factors with forced lubrication the rate of wear on the main journal bearings is always very much more rapid than on the crankpin bearings, because the crankpin bearings are always better served with oil since it is flung out to them by centrifugal force.

Oscillating Bearings.—When the motion is oscillating only, the wear on the members is no longer uniform; this objection can, however, usually be overcome by allowing the harder member to float freely; thus, in the case of a gudgeon, if this is fixed either to the piston or the connecting-rod, local wear will take place, but if allowed to float freely in bearings, both in the connecting-rod and piston, local wear on the pin can be avoided; further, a much heavier load can be carried, because the rubbing velocity between any of the members is halved.

From the above considerations, it is clear that, other things being equal:

(1) The friction of a bearing, when freely lubricated, is nearly proportional to the load factor on the bearing, and depends, though to a lesser extent, upon the nature of the surfaces—the smoother the surface the lower the friction.

(2) The rate of wear is also proportional to the load factor.

(3) When oil of higher viscosity is used, the friction is greatly increased at first; but on account of the greater amount of energy dissipated in shearing the oil film, the heat flow is greater, the temperature is therefore higher, with the result that, after running some time, the reduction in viscosity due to the higher temperature nearly compensates for the higher initial viscosity, and so the conditions as regards friction and the thickness of the oil film ultimately become nearly similar. They do not become quite similar, because owing to the higher temperature of the bearing the rate of dissipation by radiation and conduction is greater, consequently the bearing never reaches so high a temperature relative to the viscosity of the oil as when a thinner oil is used.

Types of Oils.—The oils used in internal-combustion engines

fall into two main divisions—mineral oils and those of animal or vegetable origin. "Compounded" oils, and mixtures of the two types, are also used.

Mineral oils, which are composed of various hydrocarbons, mostly of the paraffin series, are by far the most frequently used. This is partly because of their lower cost, and also on account of their chemical stability, which renders them less prone either to carbonization or to oxidation or gumming. On the other hand, they do not appear to possess the property of "oiliness" to the same extent as vegetable or animal oils. Except in cases where bearings are very severely loaded and near their limit, mineral oils are probably the most suitable, though there is some evidence that they are improved by the addition of a small percentage of other oils.

For ball or roller bearings a pure mineral oil would appear preferable, since such an oil is less liable to form corrosive acids in service.

Animal oils, such as whale and lard oils, and vegetable oils, such as rape or castor oils, are largely composed of the esters of fatty acids. Their chief virtue lies in their high "oiliness," which is of use in cases where the oil supply is necessarily limited, as in crank-case compression two-stroke engines, or where severe local over-loads, due to distortion, etc., are probable. Their defects lie in their comparative instability, which renders them liable to become gummy and acid by exposure to the air, and also causes them to carbonize more rapidly than mineral oils. They are also expensive, and, the supply being necessarily limited, would become more so if their employment became general. Their use, therefore, should be, and generally is, limited to exceptionally high-duty engines, and a few other special cases.

One such case is that of rotary aero-engines, where castor oil is used on account of its reputed non-miscibility with petrol, which in such engines is admitted through the crankcase. This might be a desirable feature in kerosene-engines, where the lubricant is contaminated by fuel condensed on the cylinder walls, though the price is very high for the general run of such duties.

Blended oils, which generally contain quite a small percentage of oils other than mineral, are in fairly wide use, and seem to be desirable in some cases, generally where the conditions are somewhat severe.

The proportion of non-mineral oil (usually castor) seems to be

too small to give the oil any great tendency to gum or carbonize, while being sufficient to appreciably increase the "oiliness."

Compounded oils are not recommended for ball or roller bearings by the manufacturers, owing to their tendency to acidify.

Carburation

The function of the carburettor is not, as is so often supposed, to gasify the fuel, but rather to provide constant proportions of finely divided liquid fuel and air under all conditions of speed or load. The gasification or vaporization of the liquid fuel takes place in part in the induction system, and in part in the cylinder of the engine.

The requirements of a good carburettor are that it shall -

(1) Provide a constant predetermined ratio of fuel and air at all speeds and at all loads, under constant conditions.

(2) Pulverize the fuel as finely as possible under all conditions.

(3) That when the throttle is opened suddenly it shall provide, momentarily, an over-rich mixture, for reasons which will be explained later.

(4) Provide an over-rich mixture for starting or running idle at very slow speeds.

(5) Be provided with automatic or at least readily controllable means of enriching the mixture throughout the whole or at least the lower part of the range, until the carburettor and induction system are fully warmed up.

(6) Be simple to adjust in the first instance and unlikely to get out of adjustment in use.

Probably no carburettor yet made conforms to all these conditions, though they are not impossible of compliance.

It is worth while to examine each of these conditions separately and to see what their compliance involves.

The first consideration, namely, that of providing a uniform mixture strength under all conditions, or of "metering," as it is generally termed, is the basic problem in carburettor design. The simplest possible expression of a carburettor is a jet to which liquid fuel is supplied at a constant level, such jet being situated in the centre of a venturi nozzle, or choke-tube, through which the whole of the air passes on its way to the engine. The depression in the choke-tube is therefore at all times a function of the I.H.P. of the engine, and this depression is relied upon to draw petrol from

the jet. Unfortunately, the laws governing the flow of liquid from a jet, and of air through a venturi throat, are not the same, for the one medium is a liquid and the other is a gas. As the speed at which the air flows through the choke-tube increases, so the flow of fuel also increases, but at a considerably greater rate, with the result that, if the proportionate sizes of jet and choke-tube diameters are chosen to give a "correct" mixture at any one speed, the mixture will be too weak at a lower speed and too rich at a higher speed, as shown approximately in fig. 32.

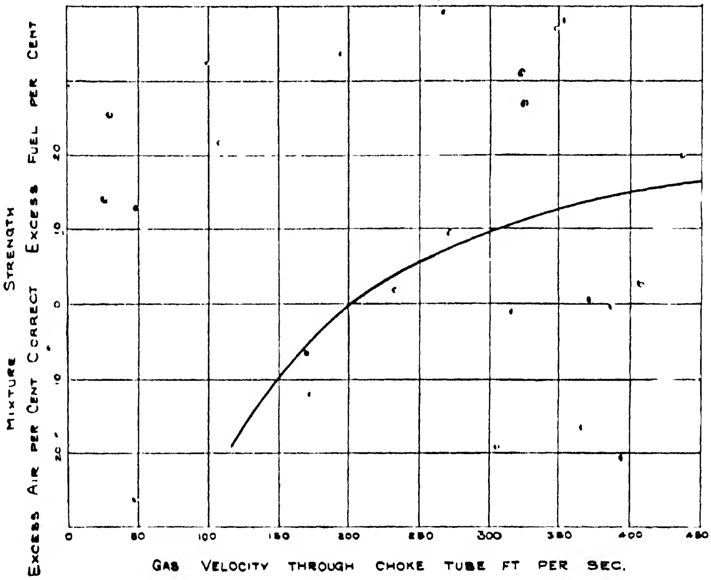


Fig 32

To this simple form of carburettor some means of compensation must therefore be provided, and for most purposes it must be automatic. There are numerous methods of compensating, but they may be divided broadly into two main groups:

(1) In which means are provided for supplying, automatically, additional air as the power output increases.

(2) In which means are provided for supplying automatically additional fuel as the power output decreases.

Intermediate between these groups are methods providing means for checking the flow of fuel through the jet by obstructing it by means of a reversed air-flow, etc.

The first group includes all such devices as automatic extra air valves operated by suction. Broadly speaking, these are not very satisfactory, because they involve the addition of a constantly moving part, which cannot readily be lubricated, and unless the movement of this part is controlled by an efficient dash-pot serious wear is liable to occur; on the other hand, if it is controlled by a dash-pot, then its movement will be somewhat sluggish, though this is not necessarily a disadvantage in view of the third requirement stated previously. In any case, however, it is always desirable to avoid the use of an additional moving part if possible.

The second group includes those carburettors in which compensation is effected by means of an additional jet fed by gravity from the float chamber and open to atmosphere; the flow of such a jet is unaffected by the depression in the choke-tube. Carburettors belonging to this group can be adjusted to give fairly accurate metering under all conditions of speed or load; and since they contain no moving parts to wear or possibly to stick, they are, in the author's opinion, to be preferred. The basic principle of this type of carburettor is illustrated in fig. 33, while fig. 34 shows approximately the rate of flow from either jet. It will be seen that, as the power output and therefore the depression in the choke-tube increases, the flow of liquid from the main jet increases rapidly, while that from the gravity-fed jet, which has a constant head of only about one and a half or two inches, remains substantially constant. Its proportional flow therefore decreases. It is obviously possible by a suitable selection of jet sizes to keep the fuel air ratio very nearly constant over a wide range of speed and load.

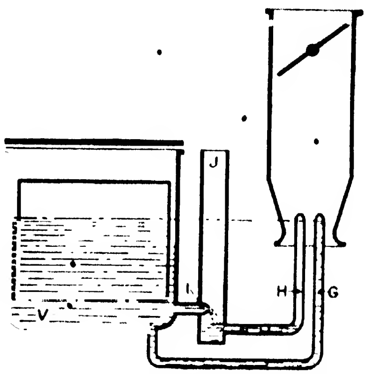


Fig 33.—Diagrammatic Arrangement, Zenith Carburettor

In addition to these general groups there are large numbers of purely mechanical devices whereby either the fuel supply, the air supply, or in some cases both, are varied mechanically by the movement of the throttle level. Such devices cannot possibly afford true compensation for all conditions of speed or load, since no change can be effected without movement of the throttle. For certain purposes, however, such, for example, as marine work in which the

torque and speed vary in a fixed relation, mechanically compensated carburettors are probably quite satisfactory. The advantages of this form are that a mechanically compensated carburettor can be made very cheaply; it has only one jet to look after and no adjustment which can be deranged. It will fulfil the requirements of a marine engine, but certainly will not give accurate metering when applied to engines in which the speed or torque may vary without movement of the throttle.

The second condition, namely, thorough pulverization, is not at all easy to comply with. It is, however, an exceedingly important

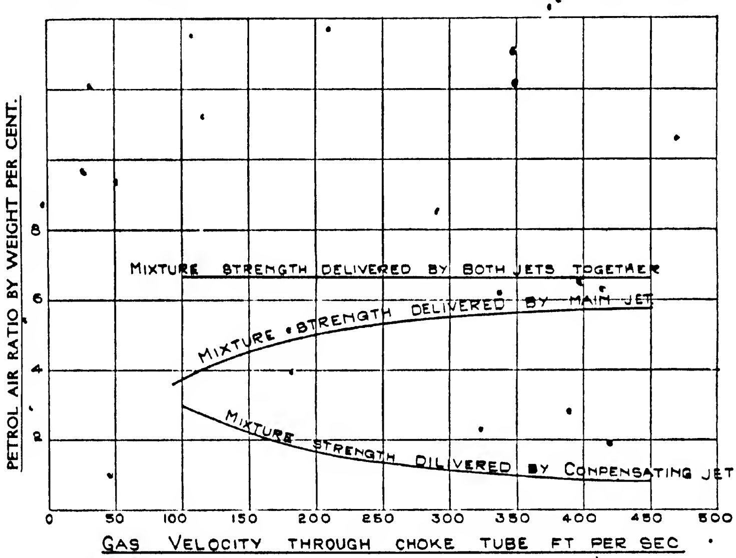


Fig. 34.—Mixture Strength delivered from Main and Compensating Jets, Zenith Type Carburettor

factor, because it is most desirable from every point of view to keep the suction temperature as low as possible. Whether the fuel enters the cylinder in a liquid or a gaseous state, it will, so long as it is finely divided, be completely evaporated after its entry to the cylinder, on coming in contact with the hot residual products therein.

In a previous chapter it has been shown that it is always very desirable to keep the suction temperature as low as possible, and to this end it is often preferable to allow the fuel to enter in a finely divided but still liquid state, and so to make use of the latent heat of evaporation of the liquid to lower the temperature in the cylinder. This, however, is possible only when the liquid is very finely

pulverized, and when the induction system is so arranged that reasonable uniformity of distribution can be obtained without too much differentiation due to the unequal inertia of air and liquid particles; if delivered in coarse drops, these will coalesce and precipitate on the walls of the induction system, finally entering the cylinder in gulps of liquid, which will never evaporate. These pass through the cylinder unburnt, while a proportion of the liquid fuel will find its way past the piston and into the crankcase, where it will contaminate the lubricant.

In most carburettors the velocity of the air past the jet is relied upon to pulverize the fuel, and, for this purpose, every effort is made to obtain a high velocity at the least possible sacrifice in power output.

Unfortunately, however, pulverization becomes of most importance when the engine is running at low speeds or reduced loads, i.e. when the quantity of air passing, and therefore its velocity, are at a minimum. With a venturi orifice of the best possible design the volumetric efficiency, and therefore the power output of the engine, will be penalized severely if the air velocity exceeds 400 ft. per second, and the author has found that to obtain a good compromise between maximum power output on the one hand and good economy on reduced loads on the other the mean velocity through the choke-tube should not exceed 400 ft. per second when four cylinders are drawing from a single carburettor, 330 ft. per second for three cylinders, and about 250 ft. per second for single cylinders, the lower velocity in the latter cases being permissible because:

(*a*) With less than four cylinders the suction is intermittent and the maximum velocity therefore considerably greater.

(*b*) The fewer the number of cylinders drawing from any one carburettor the shorter the total length of induction pipe.

To obtain better pulverization one or other of two methods may be employed:

(1) A very small choke may be used and the bulk of the air admitted elsewhere, the bulk supply of air being cut off as the load is reduced—this entails a combination of mechanical and fluid compensation.

(2) What is termed a shrouded or diffuser jet may be used in which air is drawn through the liquid to form an emulsion, which is then delivered from the jet, as employed in the Claudel carburettor, figs. 35 and 36.

The former has the advantage that it becomes possible at reduced loads to maintain the velocity not only past the jet, but also throughout a considerable proportion of the induction system. It carries with it, however, the disadvantage that the carburettor becomes somewhat complicated.

The diffuser or shrouded jet gives good pulverization at the jet itself, but, owing to the low velocity in the whole of the induction

Fig. 35.—Section, Claudel Carburettor

system at light loads or low speeds, particularly the latter, the finely divided particles are given too much opportunity to coalesce. It carries with it, however, the additional advantage that the flow of air through the fuel in the jet tends to effect a certain, though limited, measure of compensation.

For the carburation of all engines liable to sudden demands, such as all road vehicle engines and all engines under the control of a sensitive governor, it is most important that, on the sudden opening

of the throttle, the carburettor shall deliver momentarily an over-rich mixture. The reason for this is as follows:

When an engine is running light or at a very much reduced load the pressure in the induction system may be only about 5 lb. per square inch absolute. At this pressure and even at quite low temperatures almost the whole of the fuel flowing through the induction system will be evaporated and the walls of the induction passages will be dry. If now the throttle be opened suddenly the pressure in the system will at once rise from, say, 5 lb. per square inch to nearly 15 lb. per square inch absolute, while the temperature conditions will remain unaltered. Now, although the fuel may evapor-

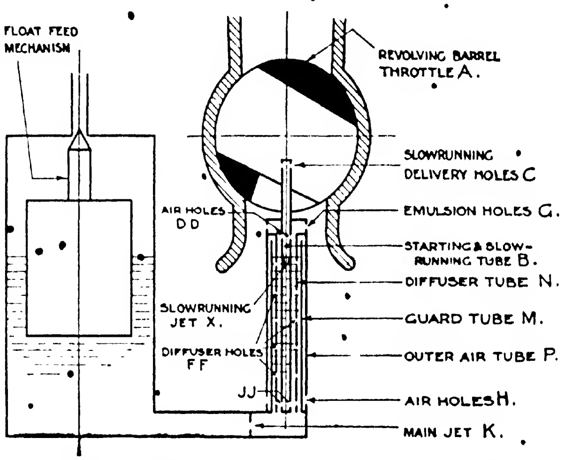


Fig. 36.—Diagrammatic Arrangement, Claudel Carburettor

ate completely when under a pressure of only 5 lb. per square inch, when the pressure is raised by nearly three times this will no longer be the case, unless the induction system be very hot. The immediate result is that the first portion of the fuel admitted after the throttle is opened deposits at once on the walls of the induction system, and, unless the mixture supplied by the carburettor is very rich, that which reaches the cylinders is far too weak to burn; this state of affairs will continue until sufficient fuel has been supplied to thoroughly wet the walls of the induction system, for when working under full throttle conditions the whole of the walls are normally lined with a layer of liquid, the thickness of which depends both on the temperature of the walls themselves and upon the velocity of flow through them. The practical effect of this is that, when the throttle

is opened suddenly after idling, the engine will splutter and backfire or even stop firing altogether for a few revolutions; if now the throttle be closed again, the half-formed wet layer will immediately re-evaporate and the engine will run steadily again; by repeatedly opening and closing the throttle the necessary wet layer can be built up gradually. To obviate this difficulty one or other of three expedients must be adopted:

(1) The carburettor must be set to deliver, at all times, an over-rich mixture.

(2) The walls of the induction system must be maintained at so high a temperature that little if any liquid fuel can lie upon them, even at atmospheric density.

(3) The carburettor must be provided with some means whereby a little liquid fuel is stored up when running idle, and delivered to the induction system immediately the throttle is opened.

The first of these may be dismissed at once as altogether too extravagant; yet it is the expedient most commonly adopted, as is evidenced by the fact that nine motor-car drivers out of ten will complain that they cannot combine good economy and acceleration. The second method can, at best, be but a partial remedy only, though a certain amount of pre-heating is essential in the case of present-day petrols, whose mean volatility is low and most of which have a final boiling point of over 400° F. To raise the induction system, however, to such a temperature as will prevent entirely any condensation even of the highest boiling fractions is practically out of the question, and would, in any event, so reduce the power output, increase the tendency to detonate, and, by raising the whole cycle temperature, so lower the efficiency of the engine, as to be quite outside the range of practical politics.

The third method, namely, the momentary supply of an over-rich mixture, meets the case satisfactorily; it costs nothing in power output, and permits both of working normally with the most economical mixture strength, and of reducing the heat input to the induction system.

In the case of carburettors using a gravity-fed compensating jet, this condition can readily be met by providing a well having a capacity sufficient to supply a 100 per cent excess of fuel for, say, 3 or 4 cycles, and fed from the compensating jet. When running on full throttle this well is normally dry, but when idling the well fills up to nearly the level in the float chamber. So soon as the throttle is opened the sudden depression caused by the inflow of air to the

induction system draws the whole of the contents of the well into the induction system and thus provides, momentarily, an over-rich mixture.

When a shrouded or diffuser jet is used the same effect can be brought about by providing in the annular passage around the jet sufficient capacity to meet this instantaneous demand.

Both the Zenith and the Claudel carburettors cater for this condition, the former by means of a well fed by gravity, as shown in fig. 37, and the latter by the use of a diffuser jet with large capacity.

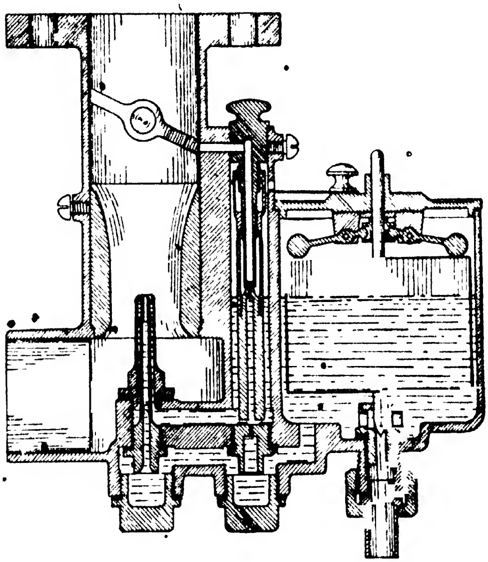


Fig 37.—Sectional Arrangement of Zenith Carburettor, showing details of Pilot Jet

Neither, however, provides, in the author's opinion, sufficient capacity to meet the case of a four-cylinder engine with a relatively cool induction system.

The fourth condition, namely, that it shall provide an over-rich mixture for starting, is met, in nearly all cases, by the provision of a separate, or pilot jet, and some provision is usually made whereby this jet drops out of action so soon as the engine has attained sufficient speed to bring the main jets into operation. In those carburettors using a gravity-fed compensating jet, the pilot is fed from the compensating jet and automatically drops out of action so soon as the level in the well falls beyond a certain point.

To start from cold, i.e. 60° F., on ordinary commercial petrol,

it is necessary to provide a minimum mixture strength from three to four times over-rich, depending upon the vapour tension of the fuel and upon the actual temperature of the mechanism. For the purpose of starting merely, there is no objection to using an excessively over-rich mixture, provided that its supply is discontinued so soon as the engine is under way. When the pilot jet draws from a well fed by the compensating jet and is but slightly submerged, as in the Zenith or Claudel carburettor, it will deliver an excessively rich mixture only when the level is at a maximum, and this again can occur only when the engine is at rest, for the very minimum running speed will suffice so to lower the level in the pilot jet chamber as to cut this jet either partially or completely out of action. When, however, the pilot jet is fed direct from the float chamber, it is necessary, in order to obtain a sufficiently rich mixture, either to flood the carburettor and so both raise artificially the level in the pilot jet and at the same time expose for evaporation a considerable surface of liquid petrol in and around the air intake to the carburettor, or partially to close the air intake to the carburettor and thereby subject the main jet to excessive suction.

The fifth condition, namely, adjustment of mixture strength to compensate for changes in temperature, is met only in those carburettors which are provided either with variable jets or with a control on the air supply. It is an important condition, but one which is generally unprovided for; indeed, it is not at all an easy one to meet adequately. Although an engine should require the same mixture proportions, once the induction system has been wetted, whether it is hot or cold, yet it is none the less desirable to reduce slightly the size of the jet as the carburettor warms up, because the viscosity of petrol and indeed of most volatile liquid fuels varies considerably with temperature, and, with a given size of jet and a given depression, the quantity of fuel passed will increase as the temperature of the liquid rises due to the reduced viscosity. The influence of the variation in the viscosity of the liquid can be reduced to the minimum by employing always very short jets or at least a very short length of orifice of small diameter, but it cannot be eliminated entirely by such means.

In all carburettors used for aircraft, and which are therefore required to operate under wide variations both of density and temperature, it is absolutely essential to provide some means of varying bodily the mixture strength. This is effected by means of what are termed altitude controls. Altitude controls take several different

forms, but the most usual is that in which the float chamber is hermetically sealed and the air space in it connected to the carburettor at two points, one below the choke-tube and the other between the choke-tube and the throttle. Between these two points there is a considerable difference in pressure, due to the resistance offered by the choke-tube. A control cock is fitted in the passage connecting above the choke-tube, and this passage is made very much larger, the other being in the nature of a small permanent leak only. When the control cock is closed the pressure in the float chamber is equal to that of the outside air, since it is balanced by the permanent leak, but on opening the control cock the pressure in the float chamber falls to something between that ruling in the passage above the choke-tube and that of the outside air, depending upon the respective areas of the connecting passage-ways. As the pressure in the float chamber is reduced, so the level in the jet falls, and less liquid is delivered from it. By adjustment, therefore, of the control cock it is possible to lower the level of liquid in the jet to any desired extent, and so to vary the mixture strength. This method, and the several variations of it, serve admirably for aircraft when the engine is driving a propeller and the torque-speed characteristic is therefore definite throughout the range; but it is in the writer's opinion somewhat doubtful whether it forms a suitable means of control for such purposes as motor vehicles, owing to its influence either on the gravity-fed compensating jet or upon the diffuser when such jets are used. For carburettors with mechanical compensation it would no doubt work admirably.

The sixth condition, namely, that it shall be simple to adjust in the first instance, and unlikely to get out of adjustment in use, is a very important one indeed, because in practice it is often very difficult to determine whether the metering is correct or not too weak a mixture, either throughout, or at any point in the range, shows itself at once by backfiring into the induction system, but it is not always so easy to make sure that the mixture is not too rich. Carburettors in which a gravity-fed compensating jet is used are always rather difficult to adjust accurately in the first place unless the engine is run under conditions wherein its fuel consumption, etc., can readily be determined at all loads and speeds, as on the test bench. On the other hand, such carburettors, once adjusted, will remain in adjustment for all time.

Carburettors with moving parts controlling the compensation are, on the whole, much easier to adjust in the first instance, but are

liable, owing to wear, leakage, etc., to lose their adjustment. Finally, there are, on the market, several carburettors which, though they do not under any circumstances give correct metering, are none the less so easy to adjust to give a fair average proportionality, that in unskilled hands they are often found to yield results as good as or better than those of more scientific design.

As stated previously, there is, so far as the author is aware, no single carburettor which conforms to all the conditions he has laid down; but despite this, a good modern carburettor when properly adjusted is a remarkably efficient piece of apparatus. Though often much abused, the fault more often lies with the distribution system than with the carburettor itself.

It must be remembered always that the carburettor and the distribution system are closely interdependent. When the distribution system, either by its large exposed surface, the low velocity maintained in it, or its low temperature, is such as to encourage precipitation, then it is desirable to use a carburettor which will give very thorough pulverisation, even at the expense of some loss by wire-drawing, and which will supply a large excess of fuel when the throttle is opened suddenly. On the other hand, with a different distribution system quite other characteristics may be desirable. It is quite wrong to suppose that any standard carburettor may be tacked on to any existing distribution system without regard to the particular characteristics of either.

Ignition.—By a process of elimination ignition systems for high-speed internal-combustion engines have been reduced to two main types:

(1) The high-tension magneto.

(2) The high-tension coil and battery system.

The former was in almost universal use until a few years ago; the latter is a reversion to an earlier type, and has come into prominence again because nearly all vehicles and aircraft are now equipped with electric-lighting sets, so that an ample supply of low-tension current is always available.

Intensity of Spark.—Although when working on full throttle with a "correct" or slightly over-rich mixture the intensity of the spark is of very little importance, yet on reduced loads or weak mixtures the intensity of the spark plays a very prominent part. It has been found experimentally that when the nature, conditions, or consistency of the fuel/air mixture are such as to yield very rapid burning, then the intensity of the spark plays but little part; and in

fact, under these conditions no difference in power or efficiency could be detected after reducing the intensity of the spark until the reduction was such that the spark failed altogether to jump across the points. On the other hand, when, owing to the presence of an excess of exhaust products, or to the use of a weak mixture, or other cause, the rate of burning was reduced, the intensity of the spark became a matter of importance. In other words, so far as maximum power output alone is concerned, the intensity of the spark does not play any serious part; it has, however, a considerable influence upon the maximum efficiency attainable, since it governs, to some extent, both the range of burning on the weak side and the rate of burning when throttled.

In the ordinary high-tension magneto, the intensity of the spark is at a maximum only at the point of maximum flux, and is therefore reduced when the time of ignition is either advanced or retarded beyond this point. In the coil and battery system, on the other hand, the intensity remains the same irrespective of the time of ignition. This is a substantial argument in favour of the latter system.

In the past, the coil and battery system was superseded by the high-tension magneto on the score of reliability for two reasons:

(1) Because there was no charging dynamo available, with the result that the battery was not kept fully charged and was frequently allowed to run down, with consequent complete failure of the ignition system.

(2) The early low-tension contact-breakers were generally ill designed and badly made, and gave continual trouble.

When the high-tension magneto first appeared on the market it had two substantial advantages over its rival system: namely, a constant supply of low-tension current, and a really well-designed and well-made low-tension contact-breaker.

These advantages no longer exist to-day, for nearly every high-speed engine is equipped with a charging dynamo for lighting and often for starting also, and thoroughly well-designed and well-made low-tension breakers can now be obtained.

It is not proposed to discuss the details of either system, for these are now well known and can be found in any text-book on the subject.

Sparking Plugs.—Probably no part of an internal-combustion engine is more complained of and abused than the sparking plug, though the complaints levelled against it are often unjust. It is

generally complained either that sparking plugs oil up and so become inoperative, or that they give rise to pre-ignition; but the fault quite as often lies in the choice of an unsuitable plug for the engine or in defective piston design, with the result that too much oil passes into the combustion chamber.

In any type of sparking plug there is a limited range of temperature between which the points will be sufficiently hot to burn off any oil which may be deposited upon them, and sufficiently cool to avoid pre-ignition.

In those engines in which, owing to defective piston design, the quantity of oil passing the piston is excessive, it is necessary, in order to burn off the oil, to employ a type of plug with thin points which will keep hot when running on reduced loads. In this connection it should be remembered that, in a throttle-controlled engine, the actual temperature of the working fluid is nearly the same at any throttle opening, and that it is the total quantity of heat and not the temperature which varies. If the plug points are thin and their facilities for getting rid of the heat are poor, they will attain a temperature corresponding to the mean temperature of the cycle, irrespective of the quantity of heat liberated. If, on the other hand, they are provided with good facilities for getting rid of the heat imparted to them, then their temperature will be governed rather by the quantity of heat, but will at all times be somewhat lower. In other words, in plugs with long thin points the points will always be hot but will remain at much the same temperature at any load; while in plugs with short, thick points and good facilities for getting rid of their heat, the points will keep cooler at all loads, but their temperature will vary over a wider range as the load is varied. From the point of view of maintaining an equable temperature at all loads, comparatively long, thin points are preferable, and so long as the compression is low and there is no detonation to increase the temperature and rate of heat flow they will not give rise to pre-ignition. But when the compression ratio is high and the proportion of diluent is therefore small, pre-ignition will occur the more readily; moreover, under these circumstances detonation is the more liable to occur, and this, as has been shown previously, will give rise to overheating of the plug points. For such engines, therefore, it is necessary to use a type of plug whose points will keep as cool as possible. When, by careful design, the flow of lubricating oil to the combustion chamber is reduced to the lowest limit, it becomes possible to use a "cool" plug without trouble from oiling up, and it is then preferable

to do so, for the thick points naturally last longer, and there is less risk of overheating from momentary detonation or other causes.

For low-compression engines or for engines whose duty is comparatively light a "hot" plug may be used with advantage, more especially if they have any tendency to pass an excess of oil; while for high-compression engines a "cool" plug must be used, and the tendency to oil up must be overcome by adequate piston design. No single plug can at present be made to suit a high-compression engine which passes excess oil into the combustion chamber, but the remedy lies in the design of the engine, and it is not fair to abuse the sparking plug because it is being called upon to meet conditions outside its legitimate range.

CHAPTER VI

IECHANICAL DESIGN

In the design of an internal-combustion engine 'as in all creative work of this nature, the æsthetic side must not be overlooked. In the first place, beauty of form and of proportions is in itself an admirable guide to mechanical correctness; for mankind has come to regard as beautiful that which is mechanically correct, whether it be in nature, in architecture, or in engineering.

In general, beauty and efficiency—in the widest sense of the term—are synonymous, and the appeal of any design to the æsthetic sense is often as reliable a guide as is a mathematical analysis of its mechanical features. Again, the æsthetic side makes a powerful though an unconscious appeal to the user, whose artistic sense, mute and inarticulate though it may be, will always be roused.

The designer's first aim should be to ensure that the products of his work will receive the care and even affection which he hopes will be bestowed upon them by their users. To this end he should make an appeal to them through their artistic sense rather than to fads or fashion, for the former is innate in all mankind, while the latter may vary widely.

There is a prevalent but quite erroneous belief that the reliability and even the efficiency of an engine are, to a large extent, a function of the actual number of parts it contains. Speaking generally, there can be no greater fallacy. While it is obvious that the number of parts should be kept down to the minimum compatible with efficiency and mechanical correctness, this can very easily be overdone. Fewness of parts too often denotes excess of compromise. All design must necessarily be based on compromise, and it is upon the soundness of judgement by which the compromise is arrived at that the success of an engine ultimately depends. In an internal-combustion engine many of the parts are subjected to complicated stresses, both heat stresses and pressure stresses; and when, by multiplying

the number of parts, such stresses can be reduced or dealt with separately, this should be done unhesitatingly. No member should be subjected to compound stresses if by the provision of additional members these can be split up; for example, when a member is subjected to combined torsion and bending it is preferable, where possible, to replace it by two separate members, one designed to deal with the bending alone and free from torsion, and another subject to torsion only and free from bending. To do this may involve the introduction of perhaps six or eight times as many parts in this particular piece of mechanism, yet the safety and reliability will be many times greater. Again, in the design of an engine the problem often arises of carrying a shaft in two bearings whose perfect alignment with each other cannot be ensured in machining or is liable to disturbance in use owing to distortion, &c. In such cases the choice lies between fitting a fairly flexible shaft which will accommodate itself by flexure to slight errors in alignment, or the provision of double universal joints between the two bearings. In the former case the shaft is liable to fail ultimately from fatigue through constant flexure, however slight, while the bearings are liable to give trouble, and in any case the friction will be much greater. In the latter case safety and the minimum of friction are ensured, but at the cost of several extra parts. A choice of this nature confronts every designer almost daily, and he has to decide whether he will risk the simple expedient or resort to the more complicated one. He is generally too liable to adopt the former course on the ground of manufacturing cost, but in such case he has no right to boast that he is using fewer parts - he is doing so only because he cannot afford to use more, or because he has neither the knowledge nor the experience to appreciate the risk he is incurring. Again, in the larger sizes of engine it is often desirable to duplicate the exhaust valves, even though this may involve the duplication of the whole valve gear also. By doing so, smaller valves can be employed, and both the temperature of the valve heads and the stresses in the valve gear will be reduced, the valves will remain in good condition for a far longer time, while the margin of safety in the valve gear will be greatly increased. The net result of duplicating the exhaust valves will be that the engine will retain its efficiency for a much longer period, and will at all times be more reliable. It is better far to use 500 parts if need be, to comply with the laws of sound mechanics, than to defy them with a single part. The craving to make one single member

perform several distinct functions is often very difficult to resist, but it should be firmly controlled.

It is possible to produce an internal-combustion engine composed of only seven parts (exclusive of studs and nuts). If fewness of parts were a criterion, such an engine should sweep the board. In practice this type of engine has earned almost universal condemnation because of its unreliability and the ingenuity with which it devises different ways of going wrong. At the other end of the scale the aero-engine which, during the war, showed itself capable of running for, by far, the longest period, without overhaul, was the Rolls-Royce "Eagle," an engine which contains at least 50 per cent more parts than any other engine in the service.

That increased number of parts necessarily involves increased care and maintenance on the part of the user is a sheer fallacy. Fewness of parts saves manufacturing costs to some small extent, but it certainly confers no benefit whatever upon the user. Even to the manufacturer it does not always effect a saving, for the amount of fitting work required is often inversely proportional to the number of parts, and fitting is nowadays the most costly of all classes of work. Designers will do well to realize that "simplicity" as ordinarily understood is by no means always a virtue; in nine cases out of ten it is a positive vice.

The foregoing remarks must not be read as implying a disregard of manufacturing cost. On the contrary, manufacturing cost is generally much the most important consideration once the needs of all-round efficiency have been catered for, and by efficiency in this sense is meant not merely thermal efficiency but reliability and durability.

In days gone by, material and skilled fitting were both comparatively cheap, while tooling was a costly item. To-day, however, owing to the vast improvements in machine-tool design, tooling and grinding have become relatively cheap, while the cost of material has risen enormously and skilled fitting has become almost unobtainable. The designer therefore must accommodate himself to these altered conditions and economize material and hand-fitting wherever possible. In this connection he will often find that it is an actual economy to employ a greater number of parts if the total weight of material used is no greater and some hand-fitting is saved thereby. With well-thought-out design and accurate machine work it should be possible to eliminate hand-fitting almost entirely. Much of the hand-fitting which is done to-day is unnecessary and even

undesirable; the scraping-in of bearings, for example, is a custom which dates from the time when accurate machine work could not be relied upon and when designers did not realize the value of self-alignment. Now that crankshafts can be finished by grinding to within extremely close limits, and all their bearing housings can be machined at one operation, scraping is no longer necessary; indeed, no amount of hand-scraping will give so accurate or uniform a bearing as that afforded by the machining. Again, from an economical as well as from a mechanical point of view it is essential that all shaft-bearing housings shall either be machined at one operation or their alignment be ensured by spigoting. If, for any reason, neither of these is possible, then it is not only better but often even cheaper to provide universal joints and so be independent of alignment, rather than to rely upon a doubtful alignment secured by costly hand-fitting. Not only is hand-fitting expensive and unreliable, but it is also the most effective barrier against interchangeability. The old belief that a "hand-made" piece of mechanism is preferable to a "machine-made" dies hard, but the sooner it is buried the better; to-day it is an anachronism.

Design and Material. It is commonly supposed that, to be successful, a high-speed internal-combustion engine must of necessity be made from very carefully selected and highly specialized materials. While, of course, it is obvious that the higher the quality of the material the better, yet, with appropriate design, the ordinary materials of commerce will be found to give perfectly satisfactory results, and are, in fact, much more widely used than is generally supposed. It is only when the weight is very closely limited, as in the case of an aero-engine, or when an exceptionally high output is desired, that fancy materials become necessary.

The ordinary multi-cylinder high-speed engine necessarily involves the use of some very complicated castings, and the choice of material for these is, in practice, governed almost solely by foundry considerations. The material of the cylinder block and crankcase, for example, must be such that it will flow freely in the mould and yield a homogeneous casting, a consideration which generally dominates all others.

In general, surface hardness is of far more importance than tensile strength; for the necessity for extreme rigidity, which is the first principle in high-speed engine design, compels the use of such heavy scantlings in any case, that tensile strength plays a very secondary part. From the point of view of rigidity all steels are

perform several distinct functions is often very difficult to resist, but it should be firmly controlled.

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cast iron is more resonant than aluminium. In the engines which the author designed for the tanks cast iron was used for the crank-cases, but towards the latter period of the war, when light high-speed tanks were called for, aluminium was substituted. These were made to exactly the same design as the previous cast-iron cases, and were, in fact, interchangeable with them in every respect. It was noticed after this change that the engines were appreciably quieter in operation, a fact which was later confirmed by the reports of officers and drivers in charge.

There is another important consideration in favour of the use of

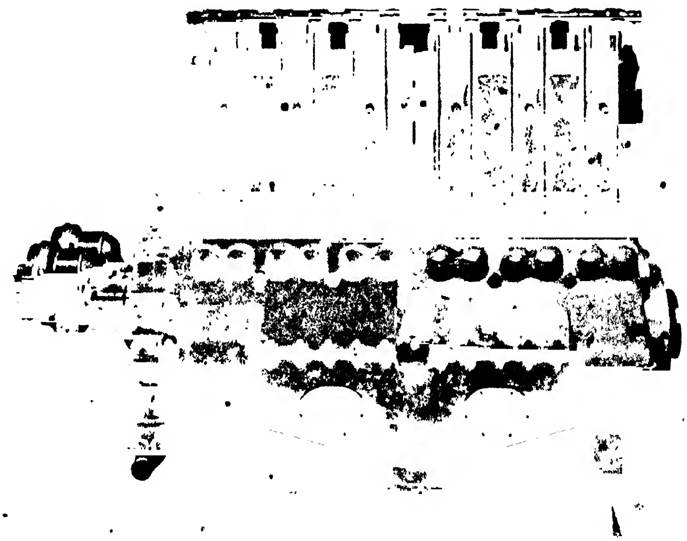

Fig. 38.—Example of High-speed 150 B.H.P. Engine

aluminium for crankcases; namely, its better heat conductivity. This is a very valuable feature, because it the more readily leads away and dissipates the heat from the crankshaft bearings, and so tends to reduce wear and the risk of bearing failures.

The principal objections to the use of aluminium are its greater cost and the difficulty in securing studs in it. The former is not so serious an objection as would appear at first sight, because—

(*a*) For complicated castings aluminium alloys are, on the whole, more reliable from a foundry point of view than cast iron; hence the proportion of wasters, often only discovered after several machining operations, is reduced.

(*b*) The cost of machining aluminium is considerably less than that of cast iron; and since in a crankcase there are usually a very considerable number of machining operations, this often outweighs the higher initial cost of the material.

The difficulty in securing studs in aluminium is a real one, but it is usually possible so to design a crankcase as to eliminate or almost eliminate the use of studs. By extending the main crankshaft bearing bolts through to the top of the crankcase and utilizing them to hold down the cylinder block, two groups of studs can be eliminated and a very sound mechanical job results. Where studs must be used they should be screwed in from to 2 to 3 diameters, and are then satisfactory, if properly fitted in the first place.

Cylinder Block.—Where a number of cylinders are cast together in one block, cast iron is almost invariably used, though recently aluminium blocks fitted with steel or cast-iron liners have been adopted for aircraft engines and for a few motor-car engines also. So far as the plain cast-iron block is concerned there is little to be said except that two considerations should be aimed at in the choice of a material: that it shall cast readily and be free from blow-holes or porous places, and that it shall be as hard as possible compatible with the first consideration. There appears to be no merit in the use of the close-grained iron so often called for in specifications, for an open grain may be just as good and is sometimes better. The main consideration is surface hardness, and this becomes particularly important when aluminium pistons are used, for these are ready enough at all times to lap the cylinder bore, and lapping can be resisted only by surface hardness. In this connection it may be mentioned that cylinders finished by grinding seldom wear so well as those which have been reamed; this appears to be due to the retention of small particles of the grinding material in the surface of the bore. These soon embed themselves in the piston, which then proceeds to lap the cylinder walls, more particularly when the piston is of aluminium alloy. The habit of grinding cylinder bores came into force at a time when it was found difficult to obtain a satisfactory finish by tooling alone; and it was soon found to be both a cheaper and a more satisfactory means of obtaining at once a reasonably accurate bore and a good finish, and so found wide favour in the eyes of manufacturers. With modern improvements in machining methods it is now possible to obtain excellent results by reaming, and this process would appear to be preferable. No doubt, with care and a suitable choice of grinding wheels, embedding of the

grinding material in the cylinder bore can be avoided, but in many instances this is not the case to-day.

In recent years it has become rather the fashion to make the whole cylinder-head detachable. This practice has many important advantages. In the first place, it simplifies the cylinder block casting very considerably and enables the cylinder to be bored straight through—a very substantial advantage. In the second, it permits of the valves being placed closer together, and so, in the case of side-valve engines, reduces the area of surface in the valve pockets and permits of the cylinder centres being reduced, both very valuable considerations. Thirdly, it eliminates the use of valve plugs, which are always an objectionable feature, since they are uncooled and are liable to leakage.

The principal objection to detachable heads is that they necessitate the making of a gas- and water-joint over a large area and the breaking of this joint in order to get at the valves or clean out the combustion chambers. By the use of suitable copper asbestos gaskets, by providing plenty of holding-down studs suitably spaced, and an ample depth of head, these objections may be overcome. Where trouble with cylinder-head joints has arisen, it can generally be traced either to lack of rigidity in the head or to insufficient or badly placed holding-down studs. Given careful design and ample rigidity, the use of detachable cylinder-heads has much to recommend it.

It is now customary to embody the induction manifold in the cylinder block casting. This certainly makes for neatness, and, by reducing the number of joints in the system, lessens the tendency to leakage; but when these points are conceded and admittedly they are important points—there is nothing else left to recommend the practice, which has three serious defects:

(1) The design of the induction system, and therefore the efficiency of distribution, must be subordinated to foundry requirements. It is seldom, if ever, possible to design an efficient distribution system which can be embodied in the cylinder block casting.

(2) With modern petrol having a final boiling point of about 400° F., jacketing of the induction system with warm water is of very little use; it merely serves to supply heat to the working fluid before its entry to the cylinder, without ensuring evaporation of the heavier fractions. In previous chapters it has been shown that the less heat supplied to the gases before their entry to the cylinder the better. To prevent precipitation some heat must be supplied,

but to be of real use it must be supplied at a high temperature. It is far better, therefore, to supply locally a small quantity of high-temperature heat just at those points where, owing to changes in velocity or direction, precipitation of liquid fuel is most liable to occur, rather than to subject the whole system to a continuous supply of low-temperature heat, which serves merely to reduce the power output without evaporating any but the lighter fractions of the fuel.

(3) When the induction system is cast in the cylinder block the

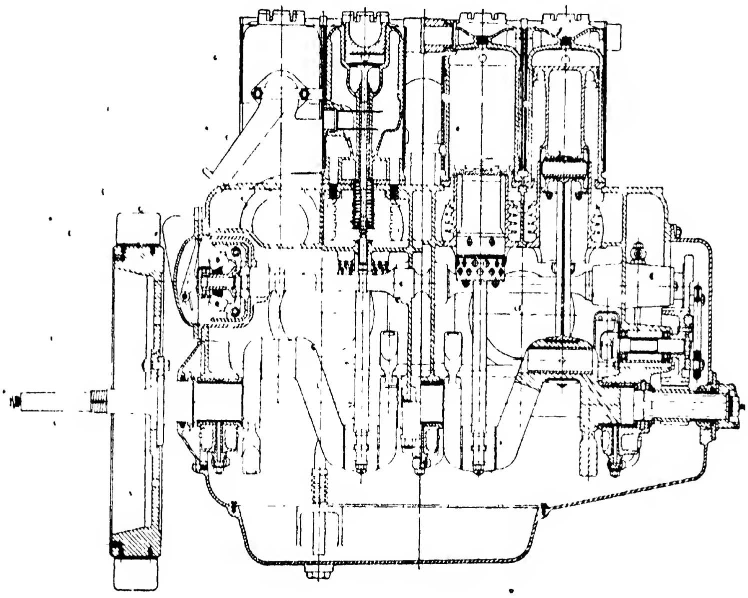

Fig. 39.—Section of Four cylinder Engine, showing Open-sided Cylinders

whole of its internal surface is rough; this greatly increases the tendency of the liquid particles of fuel to precipitate, and it increases also the frictional losses.

Where high-power output and efficiency are important considerations, therefore, the practice of casting the induction system in one with the cylinder block cannot be recommended; though for small engines in which low cost and cleanness of design and freedom from leakage play relatively a more important part, there is undoubtedly much to be said in its favour.

Apart from the orthodox multi-cylinder monobloc type, there are many other forms of cast-iron cylinder. Fig. 39 shows a fairly

common form for separate cylinders in which the sides of the water-jacket are cast open and subsequently closed with thin sheet-steel covers. This has the advantage that the casting is very simple, can be inspected internally, and the core sand thoroughly removed. Also, it permits of a number of cylinders being packed together more closely than would otherwise be possible with separate cylinders, thus both reducing the length of the crankshaft and increasing the rigidity of the crankcase, as shown in fig. 40.

Fig. 41 shows a design for large cylinders, in this instance of 100 horse-power, in which the head consists merely of a circular plug

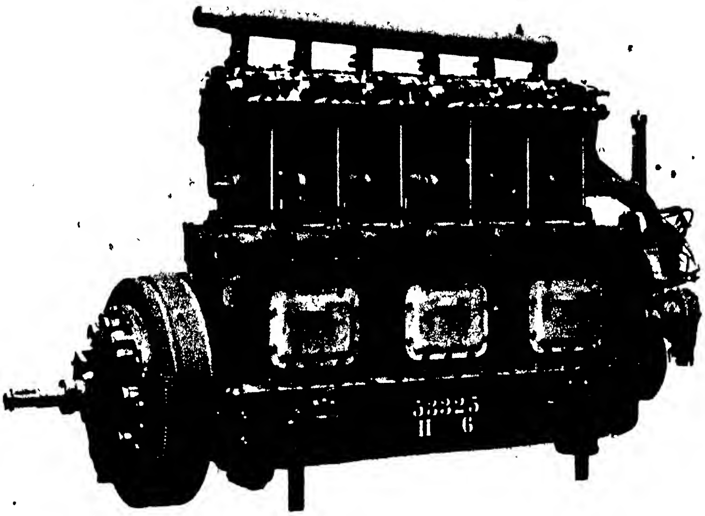

Fig. 40.—225 B.H.P. Tank Engine, showing close packing of Cylinders

carrying the valves, which can be removed very readily and with the minimum of disturbance to pipe joints, etc.

There are a great many different forms of cylinder construction in use for aircraft engines, and some of these will be dealt with in further detail when considering aero-engines. One form of cylinder block construction which the author favours for very light or high-duty engines consists in carrying the aluminium crankcase up to the cylinder head and inserting in it loose steel liners machined all over and provided with flanges at the top nipped between the top of the crankcase and the cylinder head. The lower end of each liner passes through a diaphragm in the crankcase which forms the bottom of

the water-jacket, the water-joint being made by means of a rubber ring as in gas-engine practice. This system, actual forms of which are shown in figs. 42 and 43, is of course only an adaptation of the ordinary horizontal gas-engine construction, but it has, in the author's opinion, the following advantages:—

(1) It is at once both light and inexpensive.

(2) It renders the top half of the crankcase very rigid against bending or torsion.

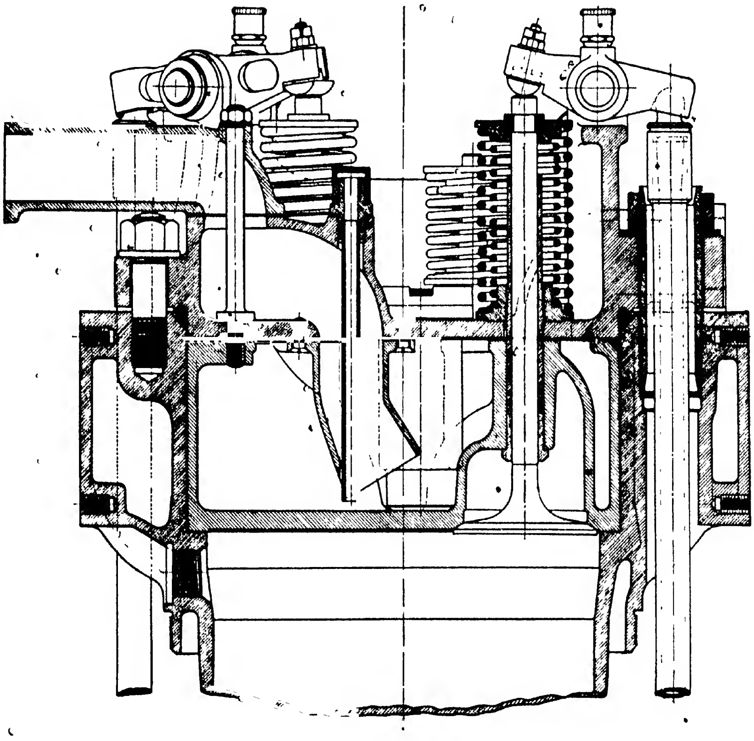

Fig. 41.—Section of Cylinder and Head, 100 B.H.P. Engine. 8¼-inch bore, 9½-inch stroke. Speed, 1150 R.P.M.

(3) It is simple and easy to cast.

(4) The water connections can be secured to the main casting, and not to light sheet-metal jackets.

(5) The liners being of very simple and symmetrical form can be made from mild steel and case-hardened.

This latter is, in the author's opinion, a very valuable consideration, for his experience has been that a case-hardened liner affords

the most perfect wearing surface possible ; thanks to its great surface hardness, it may be said to be immune from wear, even with aluminium pistons, and it can be ground out without risk of the grinding material becoming embedded in it. After a few hours' running, it assumes a mirror-like surface absolutely free from scratches, and retains this surface indefinitely. The use of case-hardened liners is, however, possible only when the design of the liner is that of a plain open-ended tube of nearly uniform thickness and perfectly symmetrical. If the design be in any way complicated, unsym-

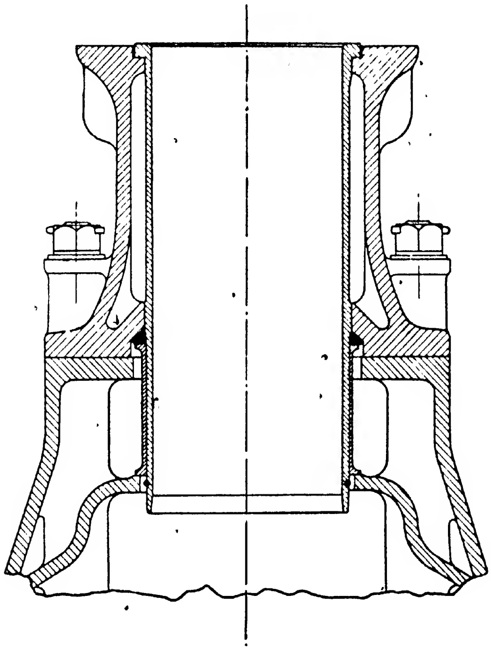

Fig. 42.—Section of Cylinder showing loose hardened Steel Liner sealed by Rubber Ring

metrical, or having widely varying thicknesses, case-hardening becomes almost out of the question on account of distortion.

Yet another form of cylinder construction which is sometimes favoured consists in casting the whole cylinder block in aluminium alloy with cast-in valve seats, and either screwed or pressed-in steel liners. This construction is open to the objection that it is always difficult to ensure good thermal contact between the liner and aluminium walls, on account of the very great difference in the coefficient of expansion of the two materials.

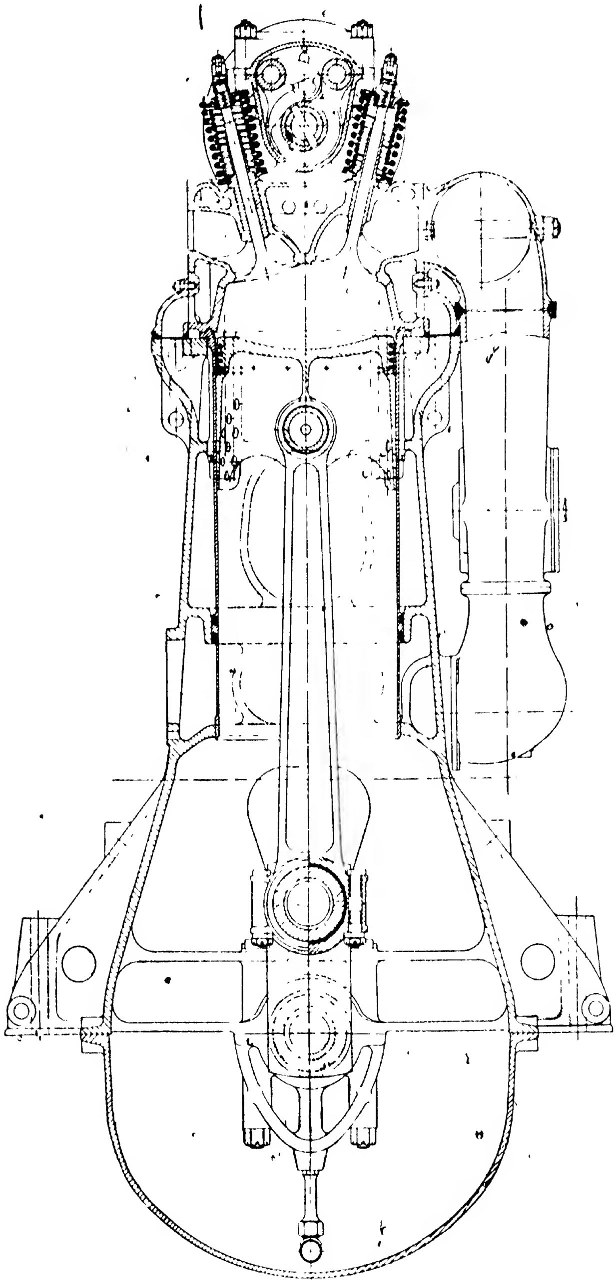

Fig. 43.—Section showing cylinder construction of large Experimental Aero-engine developing 130 B.H.P. per Cylinder. Bore 8 inch, stroke 11 inch, speed 1400 R.P.M.

A form of cylinder construction which has found great favour for aircraft engines is that in which both the cylinders themselves and the water-jackets are built up out of steel by welding. In such a construction it is customary to forge the cylinder and cylinder head as a plain thimble, to screw and weld the valve ports and valve guides into this, and finally to weld over all a light steel water-jacket pressed from thin sheet and made up in two halves. This form of construction is very light and sound, and thanks to the absence of castings, and therefore to the small proportion of scrap, it is not nearly so expensive as would appear at first sight. The chief objections to it are :

(1) It is applicable only to separate cylinders, or at least it becomes very complicated when the water-jacket embraces more than one cylinder.

(2) The water connections are attached only to light sheet-metal jackets, which are liable to crack from vibration.

(3) Freedom of design is rather limited by constructional difficulties.

As regards material for built-up steel cylinders, it would appear that a high-carbon steel about 0·4 to 0·45 per cent carbon gives good results as regards wear, but is of course rather difficult and expensive to machine. For plain open-ended liners where sufficient thickness is available there is nothing better than a straight low-carbon case-hardening steel.

For plain cast-iron cylinders a mixture should be selected which will flow freely and yield a homogeneous casting with as hard a surface as possible and regardless of " grain."

Of the three materials mentioned for cylinder liners, case-hardened steel is, in the author's experience, far and away the best from the point of view of wear and the surface it attains; cast iron, if fairly hard, and reamed, not ground, is probably second best, with high-carbon steel third. Low-carbon steel is quite useless for cylinder liners unless case-hardened. Bronze has been tried, and its use has been found to result in a higher mechanical efficiency, due probably to its better heat conductivity, with the result that the layer of oil adhering to its surface does not so readily carbonize. Its employment is, however, quite impractical, owing to its lack of surface hardness, with the result that it is lapped away with amazing rapidity.

Crankshafts.—There is always a considerable difference of opinion as to the most suitable material for crankshafts. Except

in the case of aircraft engines, tensile strength and yield point are factors which need hardly be considered, for by the time the requirements of torsional rigidity, freedom from whipping, stiffness and rigidity of crankpins and journals, and the provision of adequate bearing surface in a limited length, have all been met, it will be found that the scantlings of the shaft are such as to put all risk of failure from any source but fatigue, quite out of the range of probability. The two essential requirements are resistance to fatigue and surface hardness. In the author's opinion it is almost a waste of time to calculate out the stresses in a crankshaft, for, if the design is adequate from the points of view enumerated above, it will invariably be found that the calculable stress in the material is quite absurdly low. It must be remembered that the modulus of elasticity for all steels is substantially the same, and since it is rigidity which controls the design of a crankshaft, all steels are nearly on a par in this respect. In the author's opinion a straight carbon steel containing about 0·35 per cent carbon fulfils all the requisite conditions quite satisfactorily for most commercial purposes except for aircraft engines. These latter are excepted because—

(1) Having no flywheel, aircraft engines are less susceptible to torsional oscillations in the crankshaft, and the latter can therefore be made considerably lighter.

(2) In order to save weight in the shaft and in the design generally, a bearing is always fitted between each crankthrow, hence the tendency to whip is greatly reduced.

(3) Owing to their relatively short working life, the area of bearing surface is generally cut down in order to reduce weight.

For these reasons, the crankshaft for an aero engine may be made relatively much lighter than for other types; and, as a result, it is much more highly stressed.

Apart from the question of cost of material, the advantages of using a straight carbon steel for ordinary engines are:

(1) That every engineering establishment is familiar with the methods of forging, heat treating, and machining it.

(2) That it is very uniform in quality, and much less sensitive to errors in heat treatment.

In other words, it is decidedly more dependable than high tensile alloy steels, all of which require careful heat treatment, and if wrongly treated, are liable to be dangerous.

Fracture of a carbon steel crankshaft of reasonable design is a very rare occurrence indeed, and is generally due either to want

of adequate fillets at the crankpins and journals, or to periodic torsional vibration; either of these causes will result in eventual fracture from fatigue, no matter what the material may be. The former may be avoided by providing an ample radius, and the latter by avoiding periodic vibration, either by fitting a vibration damper or by altering the scantling of the shaft so as to raise or lower the periodic speed out of the normal running range. In the case of four-cylinder engines with reasonably light reciprocating parts, it is generally quite easy so to design the crankshaft that there shall be no torsional vibration at any speed of which the engine is capable. In the case of six-cylinder engines, however, it is by no means so easy to do this, and it then becomes desirable to employ vibration dampers or other means to check torsional vibration.

The most important consideration to-day is to reduce the rate of wear. This, as shown previously when dealing with lubrication, is a function of the surface hardness, both of the shaft itself and of the bearing material in which it runs. The harder the shaft and the softer the bearing material the better, provided that the latter will not crush. Wear appears to be due almost entirely to particles of grit embedding in the soft bearing material and so lapping the shaft. The rate at which they will lap away the shaft depends upon (1) the surface hardness; (2) the load, which governs the thickness of the oil film and therefore the distance which the particles have to bridge before coming into contact with the shaft.

In general, the load on the crankshaft bearings is so severe that it is not always possible to use a very soft bearing material, but it should be borne in mind that, other things being equal, the softer the bearing material the less will be the wear.

Balance Weights.—In four- or six-cylinder engines, the centre crankshaft journal is subjected to very severe loading on account of the cumulative centrifugal and inertia pressure from the two cranks on either side of it, since these are always in the same plane; for this reason, either much greater surface must be given to this bearing, or the crankshaft must be fitted with balance weights to counteract the centrifugal pressure from the two centre cranks. Unfortunately, if the two centre cranks are fitted with balance weights, so also must the others. The provision of balance weights, however, while it relieves the load on the journal bearings, and more particularly on the centre bearing, and also reduces the tendency to vibration of the crankcase, is very objectionable because, by adding masses with considerable inertia to the various throws of

the crank, it increases greatly the tendency to torsional vibration of the crankshaft, and at the same time tends to lower its critical speed. This may be, and often is, a serious objection to the use of balance weights, for torsional vibration of the crankshaft is much more serious in its effects and more difficult to deal with than that of the crankcase. The use of balance weights, therefore, is by no means always to be recommended; they may be of advantage in reducing wear, or they may set up severe and dangerous periodic vibration, depending upon the actual circumstances. In cases where, owing to restrictions upon length or for other reasons, the area of bearing surface of the centre bearing is limited, it sometimes becomes necessary to provide balance-weights, and in such cases their ill effects can be counteracted by the use of the Lanchester torsional vibration damper described in Vol. I under the heading of "Balancing."

There can be little doubt but that the ideal crankshaft should have case-hardened journals and pins. It is very difficult satisfactorily to case-harden a one-piece multiple-throw crankshaft on account of its tendency to buckle when quenched, except in the case of very short cranks such as are used in engines with two opposed cylinders. It is, however, in the author's opinion quite open to question whether the advantages of case-hardened bearing surfaces are not such as to justify the use of built-up cranks, with the webs shrunk on to the journals and the crankpins pressed or clamped into place. Fig. 44 shows an actual example of a crankshaft constructed in this manner for an engine of 125 B.H.P., running at a normal speed of 4000–4500 R.P.M., which has been found to give very satisfactory results. Fig. 45 shows an alternative design with the crankpins clamped into place, and roller bearings used throughout. This design is intended for an engine of normal speed and performance. The bearings on such a shaft should prove well-nigh everlasting, but in the event of wear or any accident such as the fracture of a roller, the whole crankpin, complete with its roller bearing, could be replaced quite easily. Apart from their virtual immunity from wear, ball or roller bearings have the advantage that they are not dependent upon continuous lubrication to maintain the oil film and are therefore more reliable; also, although the friction loss with plain bearings amounts, in any average design, to less than 2 per cent of the maximum power of the engine at high speeds, and at normal working temperatures, this loss is almost independent of load, and is liable actually to increase

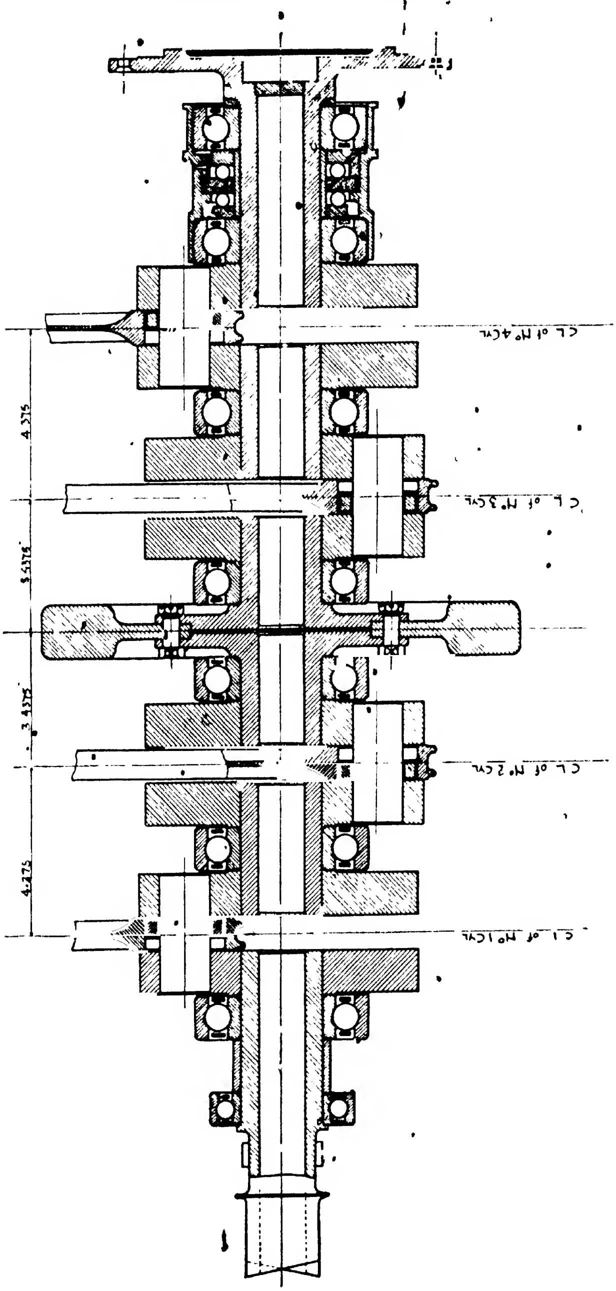

Fig. 44.—Crankshaft Assembly as used in 3 litre 1922 Vauxhall Racing Engines

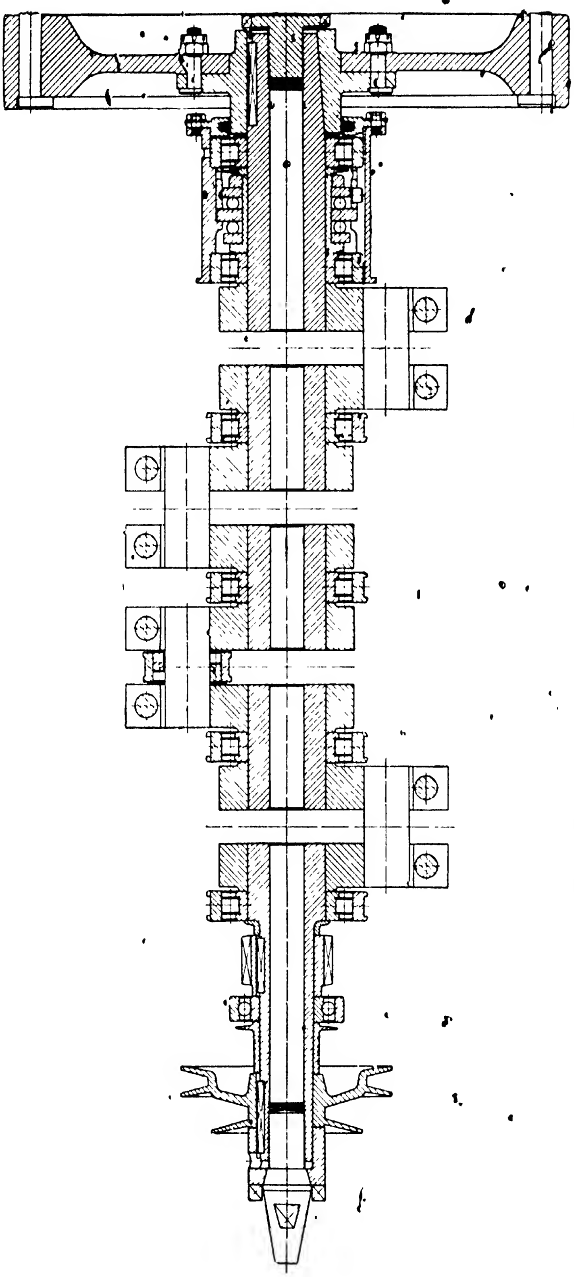

Fig. 45.—Roller-bearing built-up Crankshaft for heavy Commercial Vehicle Engine

as the speed is reduced. While, therefore, the total frictional losses of the plain bearings on a crankshaft are relatively small at high speeds and full load, at reduced speeds and loads they begin to play an important part. In the case of motor-vehicle engines which, for the bulk of their existence, operate under very light loads, and whose average load factor is only from 30 to 40 per cent, the use of a ball- or roller-bearing crankshaft would result in a very appreciable gain in fuel economy, probably about 6 to 8 per cent.

Wear of Crankshafts.—In general terms, it may be said that while the greatest wear in crankshafts generally occurs in the main journal bearings, actual failure of the bearing material is more frequent on the crankpins, even when both journal and crankpin bearings are subjected to the same load factor. The causes of this state of affairs are not far to seek. The greater liability to failure of the crankpin or connecting-rod big-end bearings as compared with the main journal bearings when both are subjected to the same mean loading per square inch is due to:

(1) The smaller heat capacity which the bearing has to draw upon.

(2) Inadequate support of the bearing in the connecting-rod, the big-end eye of which is generally lacking in rigidity.

In the event, therefore, of a temporary stoppage or slowing down of the oil circulation, the crankpin bearing will, in a given time, attain a much higher temperature than the main journal bearings, and, if the stoppage be prolonged, it will reach the critical temperature at which breakdown occurs (viz., about 360° F. for ordinary white metal), long before the main journal bearings, which have behind them the heat capacity of the whole of the crankcase.

With regard to wear, in most lubrication systems the oil is delivered first to the main journal bearings, and passes thence to the crankpin bearings, with the result that the journal bearings receive and retain most of the grit in the lubrication system, and so the more readily lap the shaft.

In general, the rate of wear appears to be almost directly proportional to the load factor on the bearing, and the surface hardness of the shaft, but is almost always most rapid at the bearing which receives its lubrication first and so retains most of the grit.

Connecting-rods.—The chief considerations in the design of a connecting-rod are:

(1) That it shall be stiff enough to resist not only bending and crushing, but also vibration.

(2) That it shall be as light as possible.

(3) That the big-end eye of the rod shall be sufficiently rigid to ensure adequate support to the bearing.

Adequately to fulfil conditions Nos. 1 and 3, means in effect that the scantlings of the rod must be such that it has an ample margin of safety so far as ultimate strength is concerned. When I. section rods are used, and this is probably the most desirable form both from the point of view of manufacture and of the disposition of the material, the section must have sufficient width to resist vibration in the plane of the crankshaft, as well as sufficient depth to resist bending—this is too often forgotten and many mysterious troubles and noises are probably due to sideways vibration of the connecting-rod.

It is clearly most important to keep the weight of the connecting-rod as low as possible consistent with fulfilling the other requirements, and in this connection it should be emphasized that it is the weight of the rod as a whole, and not that of the reciprocating end only, which has to be considered. In many cases it is quite as important to keep down the weight of the rotating as it is the reciprocating end, for while the reciprocating mass of the rod affects the balance of the engine, its rotating weight is of more importance in so far as it influences the average pressure on the crankpin and crank journal bearings due to centrifugal loading. In a six-cylinder engine in which the reciprocating parts are balanced inherently, rotating weight plays a more important part than reciprocating; on the other hand, in a four-cylinder engine in which the secondary disturbing forces due to the inertia of the reciprocating mass are cumulative, it is the reciprocating weight of the connecting-rod which must be considered first.

It is in the design of the eye of the big-end that particular care is required.

The first consideration is to obtain a uniform support for the bearing, and, to this end, not only must the eye of the rod be made as rigid as possible in itself, but also the load transmitted down the shank must be distributed over it as uniformly as possible. It is generally useless to provide a wide bearing because of the practical impossibility either of obtaining the necessary rigidity or of distributing the load over it. It is, in the author's opinion, very doubtful whether any useful bearing surface can be obtained when the width of the connecting-rod big-end bearing exceeds the diameter of the crankpin; in any case it is very desirable to splay out the outer

webs of the shank at their junction with the big-end eye, in order to distribute the loads transmitted down them.

Another frequent source of weakness in connecting-rod design is the lack of provision of sufficient abutment for the two halves of the connecting-rod bearing. The cap of the rod should be considered as though it were an arch loaded at its centre, but in tension not in compression. Viewed in this light it is obvious that unless the arch has a wide abutment it will tend to close in and so nip the crankpin at the sides. In high-speed engines it is common practice,

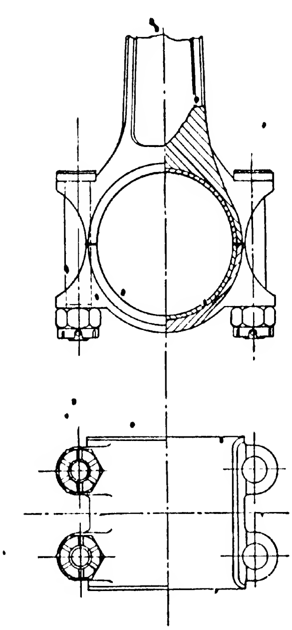

Fig. 46

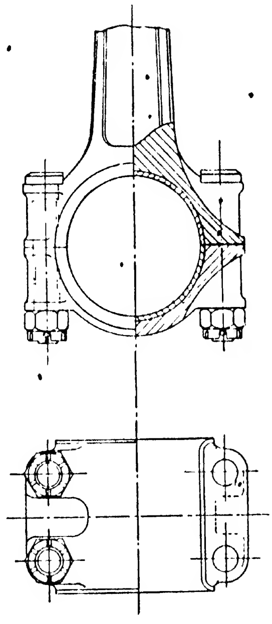

Fig. 47

in order to save weight, to cut down the width of the abutment to the lowest possible limit, and many failures of big-end bearings are directly attributable to this cause. Fig. 46 shows a design of connecting big-end used in some aero-engines which gave rise to constant failures of crankpin bearings. Fig. 47 shows the same rod after the design had been corrected, when no further trouble arose.

The problem of fitting the bearing itself in the connecting-rod big-end has always been a difficult one. There can be little doubt but that the most reliable method is to run the white metal direct

(2) That it shall be as light as possible.

(3) That the big-end eye of the rod shall be sufficiently rigid to ensure adequate support to the bearing.

Adequately to fulfil conditions Nos. 1 and 3, means in effect that the scantlings of the rod must be such that it has an ample margin of safety so far as ultimate strength is concerned. When I. section rods are used, and this is probably the most desirable form both from the point of view of manufacture and of the disposition of the material, the section must have sufficient width to resist vibration in the plane of the crankshaft. as well as sufficient depth to resist bending—this is too often forgotten and many mysterious troubles and noises are probably due to sideways vibration of the connecting-rod.

It is clearly most important to keep the weight of the connecting-rod as low as possible consistent with fulfilling the other requirements, and in this connection it should be emphasized that it is the weight of the rod as a whole, and not that of the reciprocating end only, which has to be considered. In many cases it is quite as important to keep down the weight of the rotating as it is the reciprocating end, for while the reciprocating mass of the rod affects the balance of the engine, its rotating weight is of more importance in so far as it influences the average pressure on the crankpin and crank journal bearings due to centrifugal loading. In a six-cylinder engine in which the reciprocating parts are balanced inherently, rotating weight plays a more important part than reciprocating; on the other hand, in a four-cylinder engine in which the secondary disturbing forces due to the inertia of the reciprocating mass are cumulative, it is the reciprocating weight of the connecting-rod which must be considered first.

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providing adequate support to the bearing and also on account of the deflection of the gudgeon pin itself.

So far as material for connecting-rods is concerned, the problem is somewhat the same as in the case of the crankshaft, but there is one important difference, namely, that the material of the rod is not called upon to function as a bearing surface, so that surface hardness is not required. Since to fulfil the requirements as regards rigidity necessitates automatically the provision of an ample margin of safety, there is seldom any occasion to call for high tensile alloy steels. Plain carbon steels or a mild nickel steel, which can readily be forged or stamped, will be found suitable in all but very extreme cases. Quite recently the use of special aluminium alloys has been tried for connecting-rods, and so far as the author's experience goes, has been found perfectly satisfactory in several engines in which he has used them. The advantages of aluminium connecting-rods are very important indeed, for not only is the weight of the rod reduced to less than half that of a steel rod of equal strength, but, what is perhaps even more important, its thermal conductivity is four or five times greater, with the result that the heat generated at the big-end bearing can be dissipated more readily. Experience has shown that with aluminium connecting-rods fitted with white-metal-lined bronze shells for the crankpin bearing, a considerably higher load factor can be carried, or conversely, at the same load factor, the wear on the crankpin is greatly reduced. Before one can be satisfied definitely that aluminium connecting-rods are uniformly reliable, it will be necessary to have experience of the prolonged use of many hundreds of rods, but already a considerable number have been fitted in high-speed engines and have proved consistently satisfactory over more than a year of hard service.

Gudgeon Pin.—Lack of stiffness and inadequate support of the gudgeon pin in the piston are common failings in many high-speed engines. Excessive wear and occasional seizure of the gudgeon-pin bearings are still epidemic in some engines, and although the cause is generally attributed to inadequate bearing surface or scanty lubrication, careful examination will almost invariably reveal that the real trouble is deflection of the pin causing excessive local pressures. Although the pressure on a gudgeon-pin bearing is very high, the rubbing velocity is low, and the average load factor is certainly by no means a high one. Given that the pressure is distributed uniformly over the surface of the bearing, the rate of wear and the risk of seizure in this bearing should be insignificant.

In the normal design of trunk piston, the gudgeon pin is carried in bosses which are attached only to the walls of the piston skirt, with the result that the fluid pressure is conveyed to the pin at its two extreme ends, for although the bosses themselves may be rigid enough, their attachment to the piston is by no means so. Calculation will show that, in most instances, the deflection of the gudgeon pin under the maximum fluid pressure is altogether excessive and quite sufficient to concentrate the loading on the two extreme ends of the bearing. It is clearly of the utmost importance either that the gudgeon pin shall receive its load from the piston crown at points as near its centre as the connecting-rod bearing will allow, or its diameter shall be such that, when loaded at the two extreme ends, there shall be no appreciable deflection. To conform with the latter condition is nearly impossible, since in many instances it would involve the use of a gudgeon pin of such large diameter and weight as to be prohibitive. When, however, the gudgeon pin is loaded at two points, about half the diameter of the piston apart, there is no difficulty in obtaining the necessary rigidity with a reasonable diameter. As a general rule, for engines of normal compression, the diameter of the gudgeon pin should be one quarter of that of the piston, and its true points of support one half the diameter of the piston apart. With such proportions and with a full floating gudgeon pin the working life of this bearing will be almost indefinite, even with scanty lubrication.

Unless the pin be of abnormally large diameter, it is quite useless to provide a wide bearing at the connecting-rod small end, since its provision necessitates spreading the points of support and so increases the deflection and renders the extra bearing surface valueless. In the author's opinion it is very doubtful whether any use can be made of a bearing width exceeding 0·35 of the piston diameter. Given sufficient rigidity, the wear on a gudgeon-pin bearing is extremely small, but, owing to the small angle through which the connecting-rod oscillates, it is also extremely local and tends to wear the pin oval. This tendency can be overcome by the use of a floating gudgeon pin, that is to say, by permitting the pin to turn freely both in the connecting-rod bearing and in the piston bosses, so that it will rotate slowly and wear uniformly all round its circumference. The use of a floating gudgeon pin removes also the difficulty of locating it endwise in the piston, which is a serious trouble, and becomes acute in the case of aluminium pistons, in which, owing to their large coefficient of expansion, the gudgeon pin can be tight

only when the piston is cold. Fig. 48 shows the mounting and means of location which the author has found the most satisfactory after much experience. In this design the load is transmitted to the gudgeon pin by two heavy webs extending straight down from the crown of the piston as near the centre as the connecting-rod bearing will permit. The gudgeon pin floats freely in the piston bosses, and is located endwise by means of light washers secured by wire circlips sprung into grooves near the end of the pin.

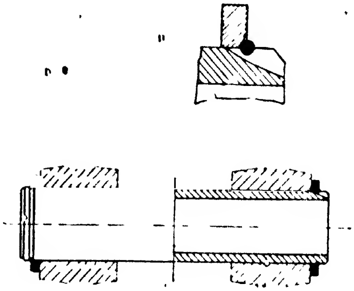

Fig. 48.—Method of locating a full floating Gudgeon Pin

For the material of the gudgeon pin one consideration, namely, surface hardness, should override all others, for if the pin is stiff enough not to deflect appreciably under load, it will be strong enough to resist fracture. From the point of view of surface hardness the best possible material is case-hardened mild steel, and a steel should be chosen which will give a glass-hard surface. In very exceptional cases air-hardened steel may be used, but this is seldom necessary.

Valves.—In the design of the valves, it is necessary always to remember that the objects in view are:

(1) To provide the freest possible entry and exit for the gases.

(2) To keep them as small as possible, consistent with the first condition.

In order to comply with these conditions it is essential to make the orifice coefficient of the valve and its surroundings as high as possible, in order that it shall pass the maximum weight of gas with the minimum of pressure difference. To this end care should be taken—

(1) That the flow of the gases on either side of the valve port is as free as possible,—so far as possible there should be no abrupt bends or changes of section on either side of the valve head.

(2) The lift should always be equal to at least one quarter and preferably even to 30 per cent of the port diameter.

(3) The angle of the valve seat should be fairly flat, 30° appearing to be about the best angle in practice.

(4) In order to give good "stream-lining" the under-side of the valve head, particularly in the case of the inlet valve, should be well radiused, the radius extending nearly to the valve seating.

The use of unduly large valves, and more particularly of large valves with a reduced lift, should always be avoided for the following reasons:—

(1) Except in very slow-speed engines the provision of relatively large valves must almost invariably be detrimental to the compactness of the combustion chamber.

(2) For a given frictional resistance, and therefore for a given volumetric efficiency and fluid pumping loss, a much higher entering gas velocity may be employed when the valve is small, has a high lift, and is well stream-lined, so that the degree of turbulence and therefore the power output and efficiency are greater.

(3) Since both the inlet and exhaust valves get rid of the bulk of their heat through their seatings, it follows that the larger the valve the higher will be its temperature. It is very important to keep the temperature both of the exhaust and inlet valves as low as possible. The former because their durability is a function of their working temperature, and the latter because the entering gases are only too ready to take up heat from the inlet valves and so penalize the volumetric efficiency. Heat picked up from the inlet valves may be regarded as purely detrimental, for it is received too late in the induction process to be of any use in assisting uniformity of distribution. Its addition from this source results merely in reducing the density of the charge and raising the whole temperature of the cycle, both of which are highly undesirable from every point of view.

(4) Large diameter valves and, in particular, exhaust valves result in unnecessarily heavy stresses on the valve-operating gear, since on full load the exhaust valves are opened against a pressure ranging from 50 to 80 lb. per square inch. That the valves should always be as light as possible consistent with mechanical strength and heat dissipation is of course obvious, but there is a tendency in many designs of high-speed engine to carry weight cutting in the valves altogether too far, with the result that stretching, distortion, and overheating are liable to occur. When the whole reciprocating mass of the valve and all its auxiliary gear down to the cam are taken into account, it will be found that the weight of the valve head alone, forms but a very small proportion of the whole, and it is generally bad policy to stint metal in the valve head and stem for the sake of the relatively small saving in weight effected thereby.

Since a valve gets rid of its heat largely through the seating, it follows that the seating should be made fairly wide in order to provide a sufficient area of contact when the valve is at rest. Very

only when the piston is cold. Fig. 48 shows the mounting and means of location which the author has found the most satisfactory after much experience. In this design the load is transmitted to the gudgeon pin by two heavy webs extending straight down from the crown of the piston as near the centre as the connecting-rod bearing will permit. The gudgeon pin floats freely in the piston bosses, and is located endwise by means of light washers secured by wire circlips sprung into grooves near the end of the pin.

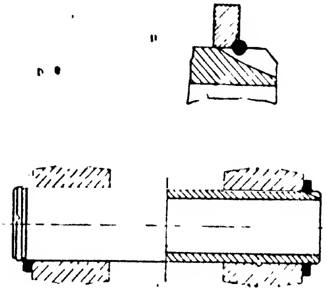

Fig. 48.—Method of locating a full floating Gudgeon Pin

For the material of the gudgeon pin one consideration, namely, surface hardness, should override all others, for if the pin is stiff enough not to deflect appreciably under load, it will be strong enough to resist fracture. From the point of view of surface hardness the best possible material is case-hardened mild steel, and a steel should be chosen which will give a glass-hard surface. In very exceptional cases air-hardened steel may be used, but this is seldom necessary.

Valves.—In the design of the valves, it is necessary always to remember that the objects in view are:

(1) To provide the freest possible entry and exit for the gases.

(2) To keep them as small as possible, consistent with the first condition.

In order to comply with these conditions it is essential to make the orifice coefficient of the valve and its surroundings as high as possible, in order that it shall pass the maximum weight of gas with the minimum of pressure difference. To this end care should be taken—

(1) That the flow of the gases on either side of the valve port is as free as possible,—so far as possible there should be no abrupt bends or changes of section on either side of the valve head.

(2) The lift should always be equal to at least one quarter and preferably even to 30 per cent of the port diameter.

(3) The angle of the valve seat should be fairly flat, 30° appearing to be about the best angle in practice.

(4) In order to give good " stream-lining " the under-side of the valve head, particularly in the case of the inlet valve, should be well radiused, the radius extending nearly to the valve seating.

it is preferable to employ two inlet and three exhaust valves, an arrangement which lends itself very conveniently to an efficient design of combustion head. Such an example is given in fig. 51,

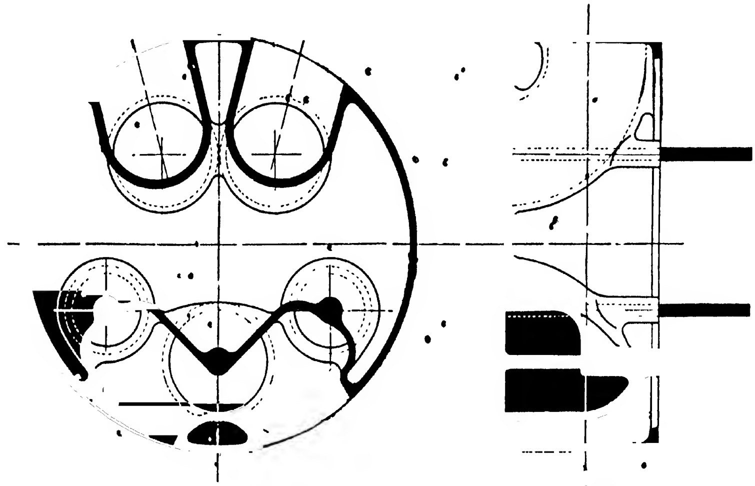

Fig 51

which shows the cylinder head and valves of an engine developing 80 B.H.P. per cylinder. In this instance two small exhaust valves and a large one are used, the former being opened some 30° earlier, in order to act as pilot valves to get rid of the bulk of the high-pressure high temperature exhaust products before the large valve opens, and so to relieve this valve and its operating mechanism.

CHAPTER VII

MECHANICAL DETAILS

Ball and Roller Bearings.—The use of ball and roller bearings in internal-combustion engines is becoming more and more extended. The great advantage of such bearings lies in—

(1) Their low coefficient of friction.

(2) Their independence as regards lubrication.

(3) Their freedom, under favourable conditions, from wear.

The disadvantages attaching to them are :—

(1) Their high first cost.

(2) Their tendency, under certain circumstances, to set up a disagreeable growling noise.

In general it may be laid down that ball or roller bearings should be used—

(1) In all places where the provision of adequate lubrication is difficult.

(2) In places where it is difficult or inconvenient to provide a sufficient surface hardness for ordinary plain bearings.

Ball or roller bearings appear to be particularly unsuitable and to give rise to noise when applied to any shaft liable to flexure, as, for example, when fitted to the main journal bearings of a very light crankshaft.

Unlike plain bearings their durability and safety are nearly independent of speed, but are dependent rather upon the *maximum* load to be carried. On the score of reliability, therefore, they show to great advantage in situations where the mean load is heavy and the rubbing velocity very high, i.e. where the " load factor," as opposed to the maximum load, is very high.

Fig. 52 shows a typical example of ball journal bearing, and fig. 53 of a similar roller bearing.

In the author's opinion a suitable situation for ball, or preferably, roller bearings, is in the connecting-rod and main crankshaft bearings because, in the first place, these are very heavily loaded, and, at the

same time, since the shaft cannot well be case-hardened all over, it is practically impossible to provide the requisite degree of surface hardness to eliminate wear. In the second place, these bearings, and these alone, of all the bearings in the engine, account for a considerable amount of friction, since their load factor is necessarily very high. The use of ball or roller bearings here will serve,

Fig. 52.—Hoffman Ball Bearing

Fig. 53.—Hoffman Roller Bearing

therefore, not only substantially to reduce the friction and so to improve the mechanical efficiency, but it will, at the same time, by reducing friction, tend to keep the lower part of the engine very much cooler—an important feature especially in large engines.

The great objection to their use for crankshaft and connecting-rod bearings is an eminently practical one, namely, the difficulty of getting them into position on a multiple throw crankshaft, since

such bearings cannot be split and must be threaded over; this entails the use of a light and lanky crankshaft, the one thing which should most sedulously be avoided. With such a crankshaft, considerable flexure is bound to take place, and this by tilting the inner race, and so partially jamming the balls or rollers gives rise both to undue wear and to the characteristic growling noise so often associated with ball bearing crankshafts. It would appear desirable, therefore, if ball or roller bearings are to be used, to build up the crankshaft, using very heavy and massive crankwebs shrunk or pressed on to the journals and so reduce as far as possible any tendency to flexure. In the case of single-cylinder engines with inside fly-wheels and built-up cranks ball or roller bearings may be used with particular advantage, for in this case the main crankshaft bearings are shielded from the maximum shock pressures by the inertia of the fly-wheels, while the use of combined fly-wheels and crankwebs permits of the fitting of a readily detachable crank-pin which can be case-hardened and ground, and being detachable, allows of the connecting-rod eye being unsplit. Fig. 54 shows an actual example of such an arrangement as applied to the "Triumph" motor-cycle engine, which has been found particularly satisfactory from every point of view.

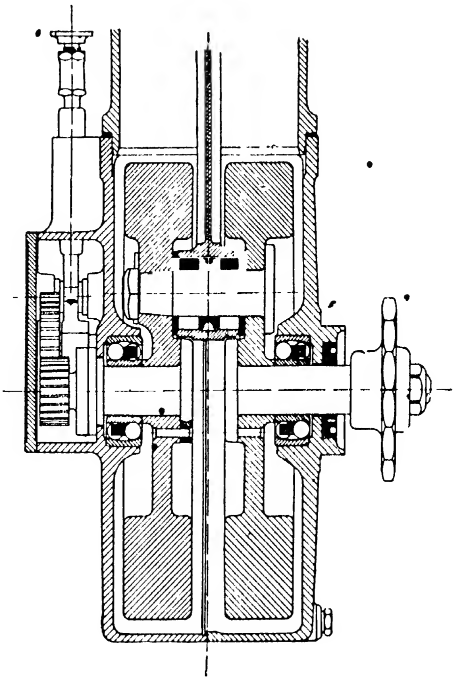

Fig. 54.—Section of "Triumph" Crank-chamber, showing Ball or Roller Bearings

Where the loads to be dealt with are heavy, as in the case of crankshaft bearings, it would appear preferable to employ roller bearings despite the objection that such bearings provide no endwise location for the outer races.

This is a serious disadvantage inherent to the roller bearing, but it is probably outweighed by other advantages.

Ball or roller bearings should always be fitted so that the inner race is secured tightly to the shaft either by being pressed on or preferably by being nipped against a shoulder; the outer race should be left a fairly free fit in its housing, and, in the case of ball bearings, should always be permitted a certain amount of sideways float. When a shaft is carried in several ball bearings it is of course essential that a single one only should be used for endwise location by the outer race, all the others being left free to accommodate themselves. Ordinary journal ball bearings are capable of dealing with a considerable amount of side thrust, and, in the author's experience, it is seldom necessary to provide thrust races to deal with casual, as opposed to continuous, end thrust. In the case of roller bearings it is of course essential to provide some independent means of dealing with end-wise location, and it is preferable always to provide ball-thrust bearings for this purpose. In cases where the alignment is uncertain the use of radial ball bearings is to be recommended, such as the Skefko bearing shown in Figs. 55 and 56.

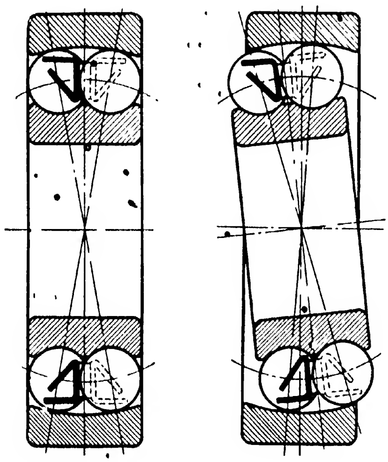

Fig. 55.—Skefko Self-aligning Ball Bearing

Such bearings, however, are liable to give trouble in erection owing to the outer housing slewing round radially when being threaded into position. The author has seen several instances in which it has been well-nigh impossible to erect certain parts of an engine on this account.

Auxiliary Drives.—In the design of any high-speed engine one of the most difficult problems, if not the most difficult, to deal with is that of driving all the auxiliary gear. This gear consists usually of the camshaft, magneto, oil-pump, water-pump, and in many cases also a fan and a dynamo. It is no easy matter to dispose all these auxiliaries in convenient and accessible positions and to drive them, each at their respective speeds, without noise or undue mechanical

complication. From the point of view of silence there can be little doubt that the use of the so-called silent chain is the best means, but it has its limitations. In the first place, it is absolutely essential to provide means for adjusting the tension of the chains if anything approaching a long working life is to be attained. Secondly, it is necessary to provide an arc of contact of the chain with its sprocket wheel of not less than 110° to 120°. This latter consideration precludes the use of a single chain embracing a number of sprocket wheels. At the very outside, a single chain can embrace only three wheels, and even this applies only when the wheels are disposed approximately at the apices of an equilateral triangle—a disposition

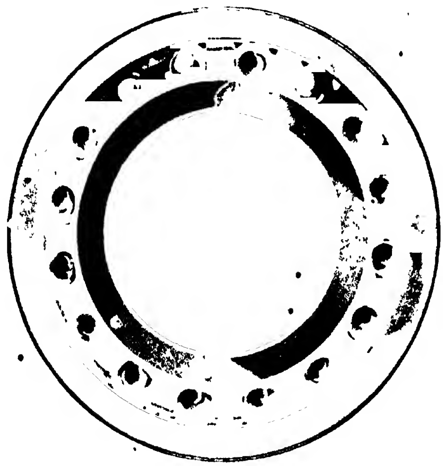

Fig. 56.—Skefko Self-aligning Ball Bearing

which is not by any means always convenient. It is therefore generally necessary to provide two chains, an arrangement which on account of their width is very bulky and cumbersome and also very expensive, for chains and sprocket wheels are at the best of times very costly items.

Fig. 57 shows an arrangement wherein a single chain is employed to drive the camshaft and magneto: in this instance the oil pump is driven directly from the camshaft, and there is no positive drive to the water-pump, fan, or dynamo, though the former, if it were required, could be driven from the shaft driving the magneto. In this arrangement provision is made for adjusting the chain by mounting the magneto shaft and wheel in an eccentric housing: the

cylindrical portion of this housing is extended through the main wall

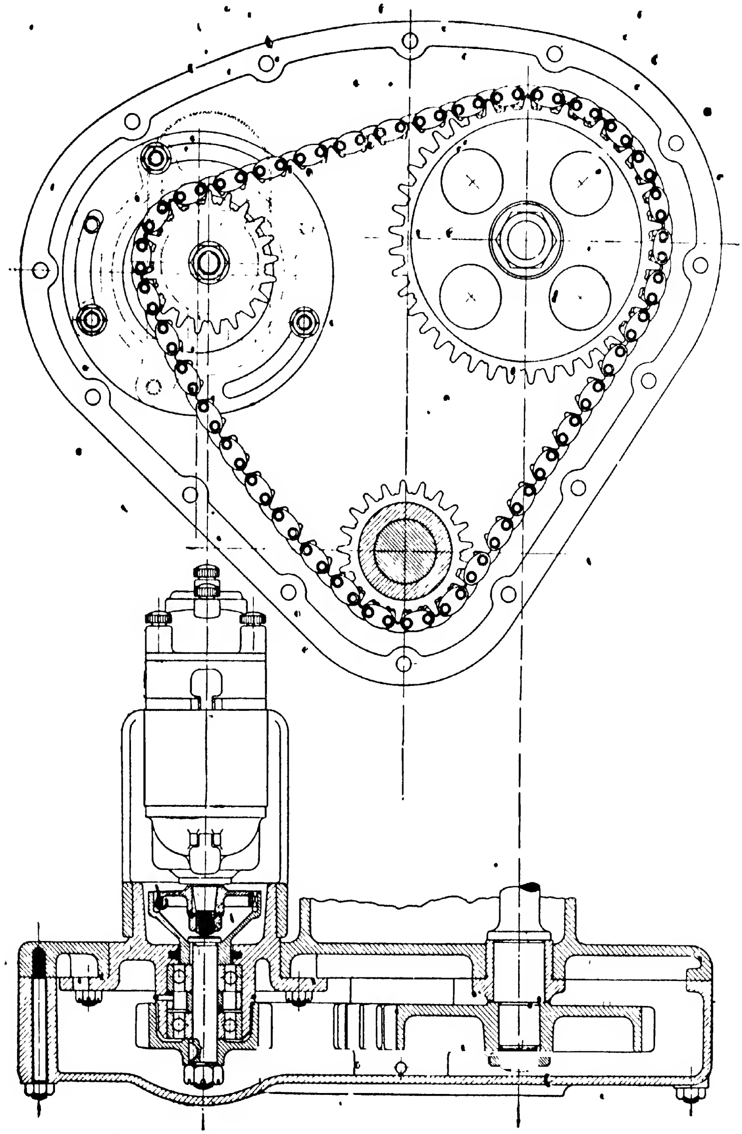

Fig. 57.—Single Chain triangulated Drive with eccentric Adjustment

of the crankcase for a length of about 2 inches, and the magneto bracket and magneto are clipped on to this projection, so that the

alignment of the magneto is entirely unaffected by any rotation of the eccentric housing.

Fig. 58 shows an arrangement for driving two camshafts, the magneto, water and oil pump. In this arrangement one chain encircles the crankshaft sprocket, a half-speed idle wheel placed immediately above, and a jockey wheel at the side, forming a nearly equilateral triangle. A second chain connects the jockey wheel with

Fig. 58.—Method of driving two Camshafts by means of a single triangular Coupling Rod

a spindle alongside, from which both the water pump and magneto are driven. The oil pump is, in this case, driven direct from the idle wheel and the two camshafts are operated by means of a triangular coupling rod, also from the idle half-speed wheel. This arrangement is very neat and compact, it has the advantage that the chains can be adjusted without disturbing any of the centres of the driven members, and is almost perfectly noiseless, thanks to the short chain centres and to the use of coupling rods for the camshafts.

The great advantage in favour of the use of chains lies in the fact that they act to some extent as dampers for the torsional oscillations of the crankshaft or, at all events, they do not transmit torsional oscillations; whereas, since it requires only a very minute change in angular velocity to reverse the loading on the gear teeth, spur gears have a tendency to chatter and scream when any period of torsional oscillation is passed through.

From the point of view of silence alone, it would be far better, where spur gears are used, to fit these at the fly-wheel end of the crankshaft where the angular velocity is virtually constant. In practice, however, there are generally serious objections to this, both on the score of accessibility and because the shaft is usually provided with a large flange for carrying the fly-wheel, formed integrally with the crankshaft, so that it is not possible to thread the crankshaft wheel into place. Such a position would, however, be vastly preferable on the score of silence, particularly in the case of six-cylinder engines, for the one serious objection to the use of spur gearing lies in its noise, and this is very largely due to variations in the angular velocity of the crankshaft at the end remote from the fly-wheel.

When spur gearing is used, it is of the utmost importance to provide that the wheels shall be correctly meshed. To ensure this it is preferable always to provide some means of initial adjustment, for it is well-nigh impossible to ensure sufficiently correct machining of the centres. When using spur gearing for the auxiliary drives, the author prefers always to mount one or more of the idle wheels in any train in a separate spider carrying the bearings for the wheel and bolted up against the face of the crankcase. This allows of some initial adjustment, for the spider can be attached by bolts or studs having clearance holes, and adjusted until the meshing is correct, when it may be finally and definitely located with a dowel pin.

Fig. 59 shows an arrangement in which two oil pumps and a camshaft are driven from a train of three spur wheels with the intermediate or idle wheel mounted on a spider as explained.

Where the number of auxiliaries is very large, it is often convenient to employ a cross-shaft driven by means of skew or spiral gearing; this arrangement is neat and compact and is much in favour for the drive of the magneto and water pump, both of which can then be disposed in a very accessible and convenient position. Such a drive is satisfactory and silent provided—

(1) That adequate provision is made for dealing with the end thrust involved and for taking up any longitudinal backlash.

(2) That ample lubrication is provided; for such gears, being of the rubbing rather than of the rolling variety, are naturally more dependent upon continuous lubrication.

There are, of course, endless possible combinations and permuta-

Fig. 59.—Photo showing Mounting of idle Spur Wheel and Auxiliary Gear

tions of auxiliary drive, but, generally speaking, the all-spur drive or the combination of spur drive and coupling rod is, in the author's opinion, to be preferred, on the grounds both of reliability and first cost; and provided suitable provision is made for adjusting the pitch correctly, and also providing that the crankshaft of the engine is sufficiently stiff to resist torsional variation, it can be made to run very silently.

When applied to motor vehicles a fan and usually a dynamo have also to be driven.

Trouble with fan drives may almost invariably be attributed to attempts to run the fan too fast. It is quite common practice to see the fan belt-driven from the crankshaft of a high-speed engine and geared to run at a speed even in excess of that of the engine. The power absorbed by a fan increases approximately as the cube of the speed, so that while at 1000 R.P.M. it may absorb only ¼ H.P., at 2000 it will absorb about 2 H.P., and at 3000 nearly 7 H.P. The belt drive provided is usually nothing like adequate to transmit 7 H.P. at 3000 R.P.M., nor would it be reasonable to expend anything approaching this horse-power in cooling the engine. The practical result is that at any speed in excess of, say, about 1500 R.P.M. the belt slips. This results in rapid wear both of the belt itself and its pulleys. In many cases, if not in most, the normal running speed of the engine is such that the fan belt is constantly slipping, which is highly undesirable, and gives rise to most of the familiar troubles with this part of the mechanism. In general it is far better so to gear the fan that it runs at a considerably lower speed than that of the engine, and so to proportion the belt drive that it will slip only when the engine is running at momentarily excessive speeds.

There are some arguments in favour of driving the dynamo by belt, for the armature of a lighting dynamo has a considerable mass and will therefore tend to run at a constant angular velocity; if it is driven positively from a member which has a varying angular velocity there is bound to be acute disagreement. For this reason the use of any form of toothed gearing for driving the dynamo is undesirable. Chain-driving may be satisfactory, because a chain, thanks to its backlash, its elasticity, and its own weight, is capable of dealing with moderate variations in angular velocity, though a belt is the best of all in this respect. Most of the troubles with belt drives for dynamos are due to inadequate size and, in some cases, to the pernicious use of a three-cornered belt drive for the dynamo and fan, a practice which cannot be too strongly condemned.

Lubrication Systems.—Broadly speaking, the various systems of lubrication may be divided into two classes; namely, those which supply the needs of lubrication alone and those which make use of the oil both as a lubricant and as a cooling agent. To the former belong all systems of trough or splash lubrication and those in which a small measured quantity of oil is fed to each bearing, while to the

latter belong all those systems in which oil is fed under pressure directly into the bearing itself. Both systems have their advantages and disadvantages, and the choice must depend upon a consideration of all the circumstances. As a broad generalization it may be said that in cases where the load factor on the bearings is high, pressure lubrication is essential because of the cooling effect obtained, while in cases where the load factor is comparatively low, splash lubrication or measured feed may be preferable, on account of the lower rate of wear, since less abrasive grit is imported into the bearing. It must be remembered that so long as oil and means of ingress are available, sufficient will always enter the bearing to maintain the oil film, whether it be applied under pressure or not, so that from the point of view of lubrication alone, as apart from cooling, the pressure system scores little or no advantage over the splash.

Pressure Lubrication.—The primary object of using pressure lubrication is to maintain a continuous circulation of a large quantity of cool oil through the bearings, in order to carry away the heat generated by friction. Bearing this in mind the objects to be aimed at are :—

(1) To circulate as much oil through the bearings as possible.

(2) To keep the oil as cool as possible.

The amount of oil that can be circulated through the bearings of an engine depends upon the pressure at which it is supplied, the clearance in the bearings, and the viscosity of the oil. It should always be remembered that the pressure in itself means nothing; it is only as a measure of the rate at which the oil is being circulated that it has any significance at all. In any pressure system, with a pump of given capacity and normal characteristics as to volume and pressure, the tighter the bearings or the higher the viscosity of the oil the greater will pressure be required to force a given quantity of oil through the bearing in a given time. If now, owing to wear in the bearings or to the use of an oil of lower viscosity, the same quantity of oil can pass through the bearings more freely, the pressure will fall, but this does not mean that either the lubrication or the cooling effect has been reduced.

On starting up, with the oil and the bearings cold, a very high pressure will naturally be required to circulate the oil through the engine, but so soon as the engine warms up the pressure will fall rapidly, due to the reduced viscosity. Under these circumstances, however, the bearings are being equally well lubricated and probably better cooled than when the oil pressure is high, since the

drop in pressure is due merely to the increased flow through the bearings. This point has been emphasized, because there is a very prevalent belief that, with pressure systems, a low oil pressure indicates defective lubrication, and it is not at all uncommon for operators to use an oil of high viscosity in order to maintain a high pressure in the system, though by doing so they are really both reducing the flow and increasing the coefficient of friction and therefore the heat generated at the bearings; in other words, defeating their own object.

In order to ensure a free circulation of oil it is desirable, with pressure feed lubrication, to keep the bearings reasonably slack. From the author's experience it would seem that a minimum clearance of about 0·0015 in. should always be permitted in all crankshaft and connecting-rod bearings when forced lubrication is employed. The practice of putting bearings up tight and allowing them to run in cannot be recommended, since it results merely in both checking the oil circulation and causing undue wear on the crankshaft itself.

Where pressure lubrication is employed and the oil is led into or near the centre of the bearing it is preferable not to use any oil grooves in the bearings, since these merely permit of the escape of oil without compelling it to circulate over the whole face of the bearing and so pick up heat from all parts; it is, however, very desirable to provide flats on the crank to help distribute the flow from the oil hole. Both on the score of reducing friction surface and eliminating the danger of nipping, it is well to relieve away at the sides of the journal bearings, though such relief should not be carried to the extreme ends and so provide a free escape for the oil.

With pressure feed lubrication it is generally desirable to use an oil of low viscosity, since this will permit of a greater quantity being circulated through the bearings in a given time and, by reducing the friction, will reduce the heat generated. So far as the author can discover, the only advantages of an oil of high viscosity are: (1) That the oil film is thicker, the surfaces are therefore kept farther apart, and there is less chance of minute particles of grit bridging the oil film and wearing the metal surfaces; (2) it appears to be less liable to work its way past the piston rings into the combustion chamber, though this latter is very doubtful; and (3) the leakage loss is reduced. From every other point of view all the advantages would appear to lie with the use of a thin oil.

Rate of Circulation.—For engines of high duty the capacity of the pump should be such that, at normal speed, about ½ gallon of oil is

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and this is liable to set up high periodicity vibrations in the oil piping and so to cause fatigue and fracture of these pipes.

The ordinary plunger pump has the advantage that its volumetric efficiency varies little with wear, and also that it has a high suction lift; but, on the other hand, the suction valve is liable to stick, either open or closed, and so put the pump out of operation; also it limits the speed at which the pump will operate (fig. 61).

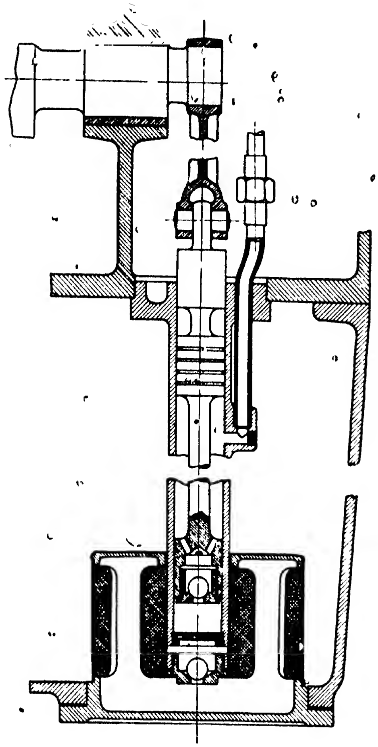

Fig. 61.—Reciprocating Oil Pump

The valveless plunger pump as shown in fig. 62 is, in the author's opinion, the most satisfactory of the three types, for it has no valves to stick or to limit the speed of operation; also it is capable of dealing with dirt, particles of felt, and the other foreign bodies with which lubricating oil is often freely supplied. When run at high speeds, as, for example, when operated direct from the crankshaft of a high-speed engine, it is necessary to provide an air vessel on the suction side in order to maintain a continuous flow in the suction pipe.

From the author's experience this type of pump is the most reliable and efficient of any he has used.

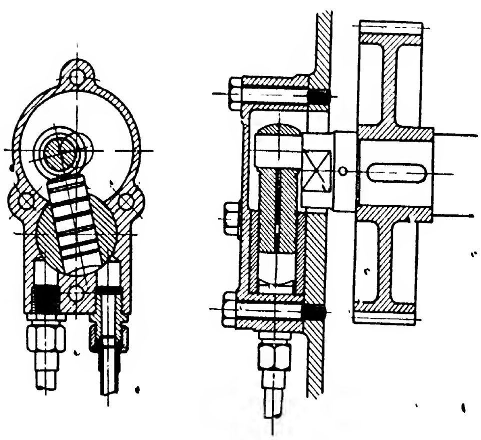

Fig. 62.—Valveless oscillating Plunger Pump

Oil Relief Valves.—In order to avoid the setting up of dangerous pressures at starting or when using thick oil or tight bearings, it is necessary in forced lubrication systems to provide a pressure relief valve, and this should be set to blow off at about 25 lb. per square inch. When gear pumps are used and the flow of oil is practically continuous, almost any type of spring-loaded relief valve will serve, but when single-acting plunger pumps are used and the flow is pulsating, the design of the relief valve requires careful consideration. A form of relief valve which the author has found to give very satisfactory results is shown in fig. 63. It consists of a spring-loaded plunger sliding in a cylindrical casing and arranged to uncover relief ports when at the end of its travel. The diameter of the plunger should be from 50 per cent to 100 per cent greater than that of the pump, and its stroke, before uncovering the relief ports, should be nearly equal to that of the pump plunger, so that it has a swept volume equal to two or three times that of the pump. It should be loaded by a long flat-rated spring and will then act as a kind of mechanical air vessel, steadying the pressure at all times, and relieving it when it exceeds any predetermined limit. With such a relief valve the pressure supply from a single-acting plunger pump is kept almost free from fluctuation and will appear practically steady on the pressure gauge.

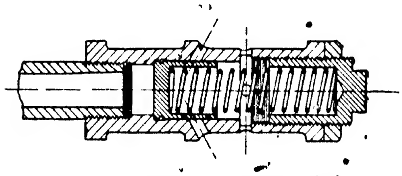
Fig. 63.—Plunger Type Oil Relief Valve

Such a relief valve should be fitted always as remote from the pump as possible in order to ensure that the oil has had access to all the bearings before reaching the relief; at the same time it is desirable that the pressure gauge connection be taken from as near the relief as possible, both in order to record a steady pressure and also to ensure that it reads the minimum pressure in the system.

Oil Filters.—The primary objection to pressure feed lubrication is that the circulation of a large quantity of oil through the bearings necessarily involves the circulation also of a large quantity of grit, and therefore tends to more rapid wear. The size of grit which causes ordinary wear, as apart from visible scoring, is such as no ordinary filter can hope to cope with, for it must be remembered that the thickness of the oil film may be of the order of one-tenth of a thousandth of an inch, and that it only requires a particle of abrasive slightly in excess of this dimension to span the oil film

and abrade the metal surface. It is obvious that no gauze or other such filter can possibly restrain the passage of particles of this size, and that the most that can be hoped for is that it will stop the larger particles, mostly of soft material, such as threads of felt from the filtration of the oil, or lumps of carbon from the pistons. The filter therefore may keep out foreign bodies which are otherwise liable to choke the oil system generally, but it cannot hope to influence the rate of wear. From the point of view of reducing wear it would appear that the best remedy is to provide ample settling space in the oil sump where the fine particles of grit can sink to the bottom undisturbed. On the whole, therefore, it would appear desirable to provide a coarse mesh filter, capable of arresting large particles which might choke up the system, and which will not readily itself become choked, and to rely upon an ample settling space for the precipitation of the fine abrasive matter.

Whether such a filter should be fitted on the suction or delivery side is still rather an open question. If on the suction side it is liable, in the event of neglect, to stop the flow of oil to the pump; if on the delivery side it is, when choked, liable to burst and deliver the whole of its collection into the bearings at one gulp. On the whole, the best compromise appears to be to fit a very coarse mesh filter capable of eliminating nuts, split pins, etc., on the suction side, and a somewhat finer mesh but substantial filter on the pressure side.

Experiments have recently been carried out with centrifugal filters or separators, and there seems a prospect that these may prove capable of eliminating some at least of the finer particles of abrasive matter and so reduce the rate of wear on the bearings. So far, however, very little is known as to their behaviour.

The system of lubrication in which a small measured quantity of oil is fed to each bearing has a good deal to recommend it for engines in which the bearings are not heavily loaded and which do not therefore require oil cooling.

The quantity of oil so fed should be the minimum required to maintain and replenish the oil film and provide a reasonable margin of safety.

The advantages of this system are:

(1) That only clean oil enters the bearings, and that only in relatively small quantities, hence the amount of grit or abrasive matter imported into the bearing is reduced to the minimum, with the result that wear also is reduced to the lowest possible limit.

(2) The amount of oil splashing about inside the crankcase is

reduced to the minimum, so that the cylinder walls cannot be over-lubricated; this is an advantage in one respect, but on the other hand it renders independent lubrication of the cylinders necessary and therefore introduces an extra complication.

The principal objection to this system, and it is a very serious objection, is that the life of each and every bearing so fed is dependent entirely on the continuous operation of the pump or other feeder, and it provides no reserve to tide over a temporary stoppage.

The use of sight-feed drip lubricators and ring oilers may be cited as the simplest expression of this form of lubrication, while, in its more developed form, it is customary to employ either a battery of small slow-moving pumps, each feeding one individual bearing, &c., or a single pump and distributing valve, the principal objection to the latter system being the relatively long interval which must elapse between replenishments.

The system of lubrication as shown in fig. 64, in which oil is fed continuously into troughs into which the connecting-rods just dip, is, in its effects, intermediate between the two foregoing. As regards wear, it has the advantage that comparatively little oil is actually forced into the bearings, and therefore the quantity of abrasive introduced is relatively small. It does not, of course, provide for any cooling by oil, since the quantity actually passing through or over the bearings is comparatively very small, but it does provide a good deal of reserve capacity in the event of any accidental stoppage of the pump. One of the difficulties, however, with this system is that it is often troublesome to provide, at all times, sufficient oil for all the crankshaft bearings without over-lubricating the cylinder walls. This difficulty can be met to a large extent by lowering the troughs, and so reducing the depth of immersion when the throttle is partially closed and the load factor on the crankshaft bearings is reduced. This can be accomplished by hinging one end of the troughs and connecting the other through suitable linkage gear with the throttle lever of the engine, as is done in the case of the Daimler sleeve valve engines.

When trough lubrication is used, it is sometimes possible to dispense with an oil pump and to use either the flywheel of the engine or a disc mounted on the crankshaft to throw oil up into a gallery from which it may be led down to the troughs by means of suitable ducts. This arrangement has, however, only a limited application, for it is not always possible to provide a disc of diameter sufficient to reach the oil, the highest level of which must be well

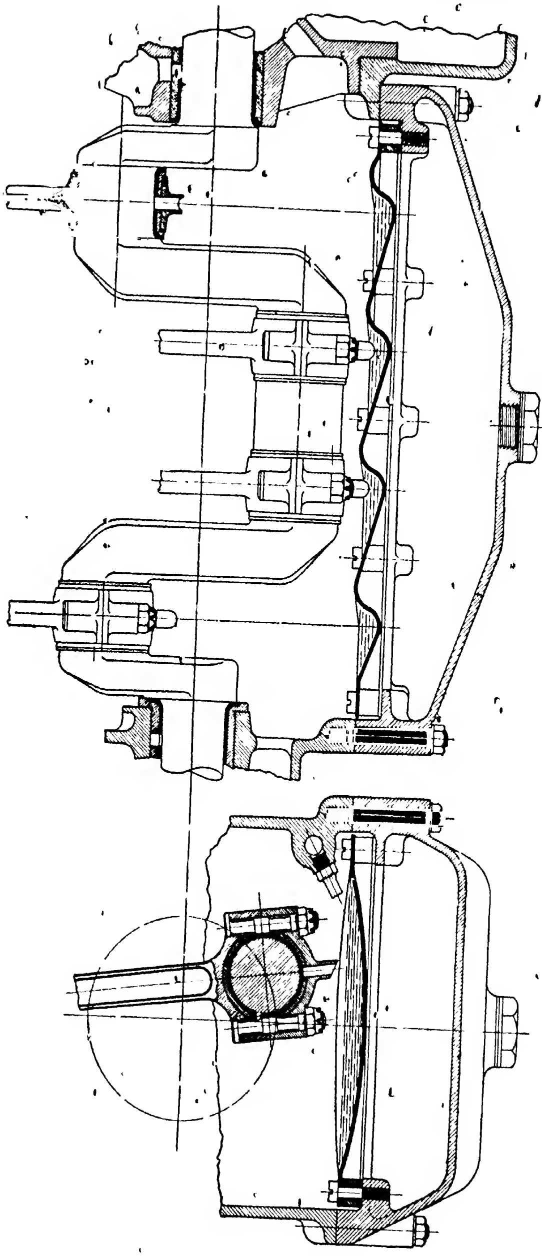

Fig. 64.—Typical Example of Constant Level Splash System

below that of the troughs. Unless there is room to fit a very large diameter disc, the permissible range of oil level becomes dangerously narrow.

To sum up: There can be no doubt but that wherever a high duty is required, involving either heavy pressures or high rubbing velocities, or both, forced lubrication alone can be relied upon, since this system provides the oil cooling so essential where high load factors have to be dealt with. The one and only objection to forced lubrication is the increased rate of wear due to the large quantity of extremely fine abrasive material introduced into the bearing. This objection can best be met by:

(1) The provision of ample settling capacity in the base chamber of the engine.

(2) The employment of materials for the bearings giving the maximum possible difference in surface hardness.

(3) The provision of centrifugal filters or separators, though not much is known as yet as to the efficacy of these devices; ordinary filtering is, however, of little use from the point of view of wear.

Measured feed lubrication is no doubt very satisfactory for lightly loaded bearings and gives the minimum of wear, but it involves the use of many oil pumps and provides no reserve capacity; as such it is rather dangerous, since the stoppage of any one pump in the group may prove disastrous. Trough lubrication is, on the whole, a good compromise for engines in which the duty is comparatively light. It tends to show a lower rate of wear than the forced system, though higher than the measured oil system. It has the advantage of providing a very fair reserve capacity, and is on the whole a fairly reliable and satisfactory system once its limitations are realized.

CHAPTER VIII

VALVES AND VALVE GEAR

The timing and operation of the valves are problems which require very careful consideration, for they are factors which have a powerful influence on the performance of the engine, and as such deserve the most careful consideration. Since the timing and opening periods must be decided upon before the cam gear can be designed, it will be well to consider this side of the problem first.

The features to be aimed at as regards the inlet valves are :

(1) To induce the maximum possible weight of charge into the cylinder at full loads.

(2) To expend the least possible energy in the process at all loads.

(3) To produce the maximum of turbulence during the period of entry.

As regards the exhaust valves, the problem is merely that of getting rid of the exhaust with the least possible back pressure and the least distress to the valve gear. So far as the exhaust valve timing is concerned, there is very little to be said except that owing to the high terminal pressure at the time when the exhaust valve is first opened, the velocity past this valve is very high indeed and the heat flow at this period very intense. The high pressure of release, however, usually provides sufficient kinetic energy to counteract the friction and inertia in the exhaust pipe, so that a high mean velocity, both through the valve opening itself and through the ports, &c., is permissible without introducing any appreciable back-pressure, provided, of course, that there is no great resistance imposed on the flow of the gases at the outer end of the exhaust pipe. On the ground of heat dissipation it is desirable, and on the grounds of back-pressure it is permissible, to use small exhaust valves and to work with a high velocity through the valve opening and ports. In practice it is perfectly satisfactory to work with a velocity through the exhaust ports 50 per cent greater than that

through the inlet, provided of course that the latter is not already excessive. In a previous chapter it has been shown that, for all-round performance, the best mean velocity through the inlet valves is in the region of 150 ft. per second. The mean velocity through the exhaust valves may, therefore, be in the region of 220 feet per second.

In the case of the exhaust valves, it is particularly desirable, from every point of view, to use small valves with a high lift, because, in the first place, an exhaust valve can only get rid of the bulk of its heat through the valve seating, for the heat carried away along the stem forms only a very small proportion of the whole. It follows therefore that the smaller the diameter of the valve and the greater its lift, the better chance there is of keeping it reasonably cool. In this connection it is necessary to emphasize that, although the maximum area of opening is attained in the case of a flat-seated valve when the lift of the valve is equal to approximately one-quarter of the port diameter, it does not in the least follow that this should represent the maximum lift, because, in the first place, the valve is fully open only for a small proportion of its total opening period, and in the second, the orifice coefficient increases rapidly as the lift is increased; or, in other words, for a given pressure difference a greater weight of gas will pass through a given area of opening when that area is provided by a small valve with a high lift rather than by a large valve with a low lift. From these arguments it will be seen that, wherever possible, the total lift of an exhaust valve should always be at least equal to, and preferably greater than, one-quarter of the diameter of the port.

Again, the exhaust valve, unlike the inlet valve, has to be lifted from its seat against a pressure which may amount to anything up to 80 lb. per square inch, and for this reason also it is obvious that the diameter should be kept as small as possible in order to reduce both noise and wear and tear.

From every point of view, therefore, it is highly desirable to use the smallest possible diameter of exhaust valve and a high lift.

As regards the timing of the exhaust valve, two factors must be taken into account. It must be opened sufficiently early to permit of the exhaust pressure falling almost to atmospheric before the return stroke of the piston commences, and it must be held open sufficiently late to permit of the residual exhaust gas escaping right up to the very end of the stroke. It is impossible to give hard and fast figures for the most suitable setting for an exhaust valve, because

this must necessarily depend upon so many variable factors, such as the mean gas velocity through the valve port and the rate of acceleration of the valve. For a mean gas velocity, however, of about 200 feet per second through the valve port and a normal rate of acceleration, the best setting, in the author's experience, is that the exhaust valve should already have travelled through 50 per cent of its total lift when the piston is at the bottom dead centre and should be still 5 per cent open when the piston is at the top centre. The actual point at which the valve leaves and returns to its seating is practically meaningless as a guide to the valve setting.

With regard to the inlet valve setting, we have to take into account a number of factors which need not concern us in the case of the exhaust valve. Also we have to be much more careful, because not only does the volumetric efficiency, and therefore the power output, of the engine depend very largely upon the inlet valve setting, but also the negative work during the suction stroke may be unnecessarily high. We have to consider how many cylinders are drawing from any one source of supply and also how far our efforts to obtain maximum power output should be subordinated to the attainment of economy on reduced loads. It will be best to consider first the conditions as they apply to a single-cylinder engine for full power, and, later, to note what modifications are necessary to meet other conditions. As in the case of the exhaust valves, we ought to use a relatively small valve with a high lift, though for different reasons. In this case we want to obtain the maximum possible turbulence at the minimum expenditure of energy, therefore we require the highest possible orifice coefficient. For full-load running, in particular, we want to get the highest possible charging efficiency, and to achieve this we want to obtain a high velocity in the valve passage during the earlier part of the suction stroke, and to make use of the kinetic energy we have acquired during this period thoroughly to fill up the cylinder towards the end of the piston's stroke. To this end we require to open the valve rather gradually at first in relation to the piston's movement, to keep it as wide open as possible towards the end of the stroke, and to shut it as quickly as possible after the end. This does not necessitate the use of an unsymmetrical cam, as may at first sight appear, but it depends rather upon the angular setting of the cam in relation to the crankshaft. To illustrate the point:

Fig. 65 shows an ordinary symmetrical constant acceleration valve

lift diagram, plotted against crankshaft degrees where 0° represents the inner centre and 180° the outward centre of the piston.

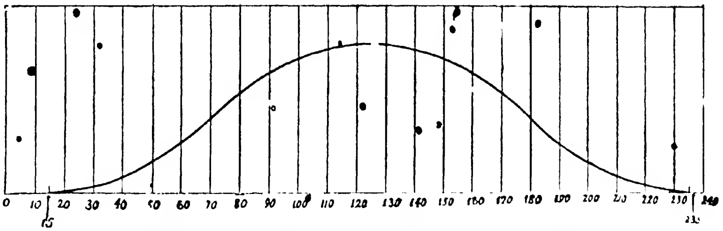

Fig. 65.—Valve Diagram, Constant Acceleration throughout

Fig. 66 shows the same diagram re-plotted in terms of piston displacement, while the dotted line represents the relative velocity of

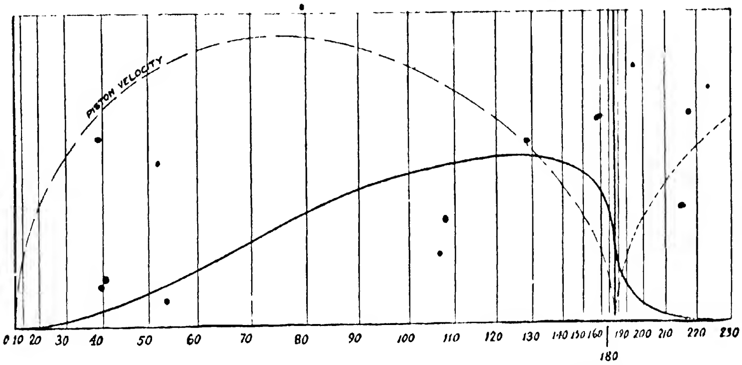

Fig. 66.—Valve Opening Diagram in relation to Piston Displacement

the piston throughout its stroke (assuming in both cases that the ratio of connecting-rod length to crankthrow is 3·6 : 1).

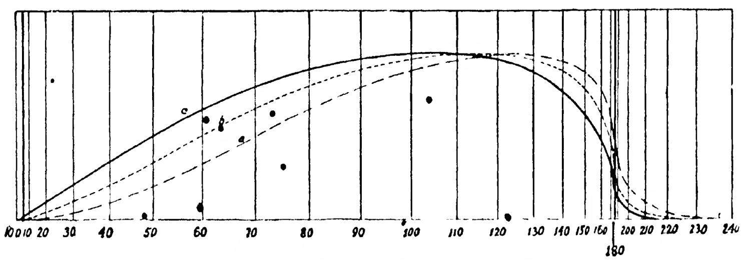

Fig. 67.—Effect of displacing Valve Opening Diagram in relation to Crankshaft

In fig. 67, *a*, *b*, *c* show the effect of displacing the angle of the camshaft by ten crankshaft degrees in each case, and illustrates

very clearly how the general form of any given valve opening diagram varies with the angular relation to the crankshaft.

Fig. 68 shows the change of velocity through the valve throughout the stroke in the case of diagram fig. 66, and assuming, for simplicity, that the pressure is constant or that the fluid is a liquid instead of a gas. From this diagram it will be seen that between the middle and the end of the stroke the velocity falls from 300 feet per second to nil, while the kinetic energy, due to this change in velocity, is made use of to pile up a static pressure in the cylinder. With careful design and with a mean gas velocity of 130 feet per second, it should be possible to pile up sufficient kinetic energy, during the earlier portion of the stroke, to overcome the frictional

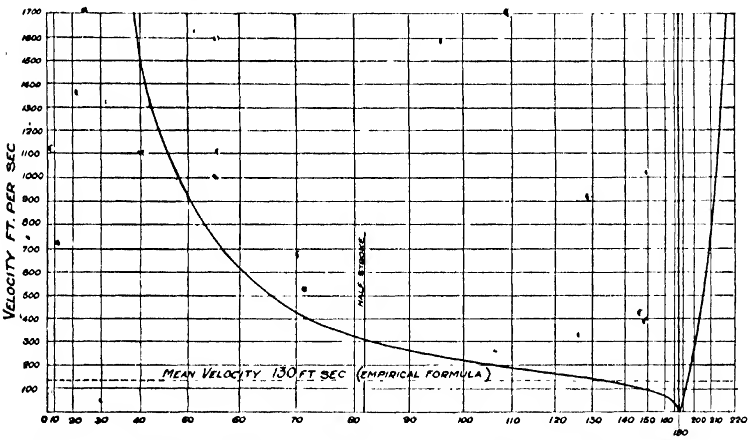

Fig. 68.—Equivalent Gas Velocity through Inlet Valve in relation to Piston Displacement

resistance of the valves, &c., and to charge the cylinder up to full atmospheric pressure by the end of the stroke; it then remains to close the valve as rapidly as possible after the bottom dead centre to avoid loss by expulsion during the early part of the compression stroke. This, then, is the setting for maximum power output, but it is not the best for fuel economy, for two reasons :

(1) Work is done by the piston in accelerating the air column during the period when it is travelling at a high velocity, and this is not returned until the piston reaches, or almost reaches, the bottom centre; consequently the pumping losses are relatively high, though, from the point of view of maximum power output, as apart from fuel efficiency, this is more than compensated for by the increased weight of the charge retained in the cylinder.

(2) When the inlet valve closes and the high velocity flow of gas towards the cylinder is stopped more or less abruptly, a reaction takes place, with the result that the air flows back through the carburettor and some fuel is liable to be blown out and wasted.

If, now, it is desirable to sacrifice maximum power for the sake of better fuel economy, it will be preferable to extend the period of opening of the inlet valve by about 20°, as shown in figs. 69 and 70, to allow it to open considerably earlier and to close a trifle later, the former in order to reduce the pumping losses and the latter to give more time to fill up the cylinder at the lower velocity. In this case,

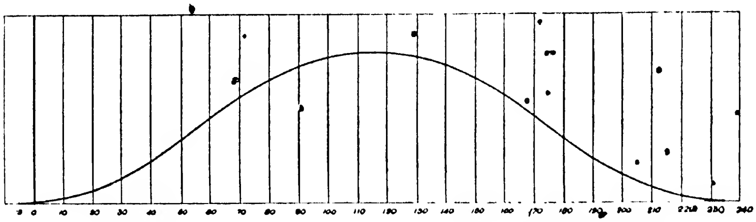

Fig 69.—Valve Opening Diagram with Period extended 20° of Crank Angle

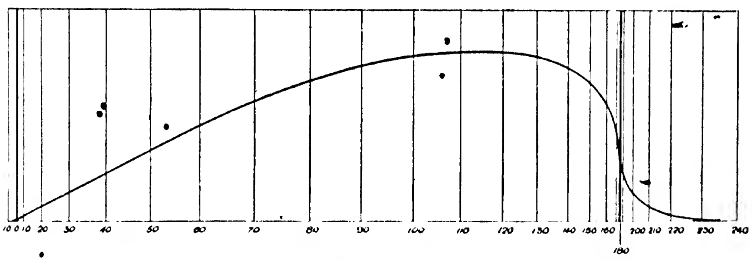

Fig .70.—Valve Opening Diagram in relation to Piston Displacement

although the opening period of the valve is 20° longer and the actual effective opening area is considerably greater, yet the maximum power output will be slightly less.

The above considerations hold good only when the carburettor is placed, as it always should be in all single-cylinder engines, reasonably close to the inlet valve port. When any considerable length of induction pipe is interposed between the carburettor and the valve port, pressure oscillations of considerable magnitude will be set up, and these will tend to surcharge the cylinder at certain speeds and to starve it at others, while at all times they will tend to increase the blow back through the carburettor.

When working on reduced loads, by throttling, we are no longer

concerned with trying to fill up the cylinder, and our aim then becomes to maintain turbulence as far as possible and to reduce the fluid pumping losses. So far as the former is concerned, we can only rely on using the smallest possible valves, upon keeping the orifice coefficient as high as we can, and upon ensuring that the gases have as unobstructed an entry to the cylinder as possible after passing the valve. Another point of importance is the position of the throttle. If this is close up to the valve port, so that there is little capacity between the throttle and the valve, then it is clear that during the idle strokes, this capacity will fill up to atmospheric pressure, or very nearly so, in which case the inlet should open early in the stroke and the suction diagram will be as shown in fig. 71; if, on the other hand, there is a considerable capacity between the throttle and the inlet port, then, at the commencement of the outward stroke, the pressure in the cylinder will be approximately atmospheric while that in the port will be considerably below atmosphere, with the result that so soon as the inlet valve opens the pressure in the cylinder will be reduced by gas flowing out through the inlet valve, and there will be some unnecessary negative work on the piston, as shown in the diagram, fig. 72. In such a case it will

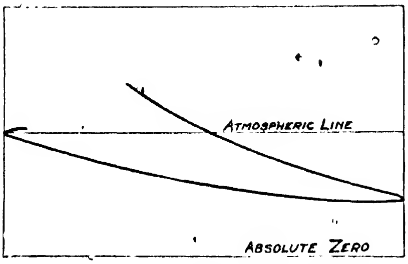

Fig. 71.—Indicator Diagram, Suction Stroke

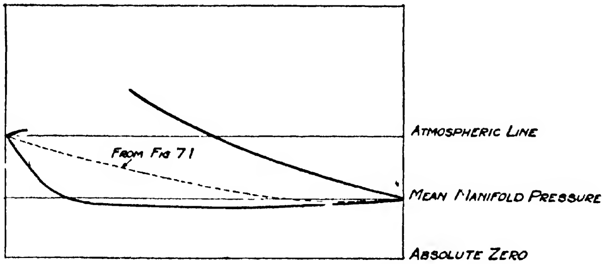

Fig. 72.—Indicator Diagram, Suction Stroke

be preferable not to open the inlet valve until the gases in the cylinder have been expanded down to a pressure corresponding to that in the port; this will give a diagram such as shown in fig. 73, which should be compared with the previous figure. It is not of course practicable

to obtain a valve timing which will be ideal for all conditions of load, and the best that can be done is to arrange the timing to suit the load at which the engine will be running for the majority of its existence. Generally speaking, for single-cylinder engines it would appear to be best to use a rather late opening inlet valve and to keep the throttle as close as possible to the valve; this will give the best results in normal working. Apart from the question of negative work, it is always desirable to reduce the capacity between the throttle and the inlet valve as far as possible, in order to subject the carburettor at reduced loads to a pulsating suction, and thus obtain better pulverization of the fuel, rather than to a continuous suck at a low velocity. Consideration will show that if the capacity between the throttle and the inlet valve were infinite the velocity through the carburettor would be continuous throughout the cycle;

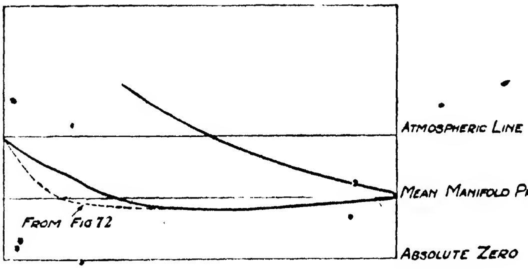

Fig. 73.—Indicator Diagram, Suction Stroke

while, on the other hand, if there were no capacity, then the mean velocity through the carburettor during the suction stroke would be just four times as great and the pulverization of the fuel correspondingly better.

The case of the single-cylinder engine is relatively simple as compared with that when several cylinders draw from one source of supply. Also it is difficult to treat the problem of valve timing and distribution separately, for they are so closely interdependent; together they form an intensely complex problem, and, probably, the least understood of all problems connected with the modern internal-combustion engine.

We will, however, assume for the time being that we are dealing with a homogeneous mixture of gas and air flowing in the induction pipe, and endeavour to see how best to deal with certain of the more common cases.

CASE 1. Two cylinders with cranks at 360° and even firing.

2. Two cylinders with cranks at 180° and consecutive firing.
3. Four cylinders with cranks at 180°.
4. Six cylinders with cranks at 120° fed by two carburettors, each distributing to a group of three cylinders.
5. Six cylinders with cranks at 120° fed by one carburettor.

So far as the exhaust valve is concerned the conditions are substantially the same in all cases, and it is the inlet valve timing alone which need be considered.

CASE 1. This is simply two single-cylinder engines operating alternately; there is no overlapping, and the problem is the same as that of the single-cylinder engine.

CASE 2. This is always a very difficult one to deal with. Probably the only satisfactory solution is to provide two separate carburettors and two exhaust pipes and so treat as two separate single-cylinder engines.

When only one carburettor is fitted from which the two cylinders draw consecutively, the best method is probably to employ very late opening inlet valves in order to avoid overlap. It is obvious that if the first piston sets up a high velocity in the induction system and relies upon the kinetic energy so acquired to fill up the cylinder at the end of the stroke, then it is fatal to allow the second inlet valve to open until the first cylinder is completely filled, for the energy acquired will be expended simply in forcing gas into the second cylinder at the commencement of its suction stroke when it is not required, while the first cylinder will be starved. So long as there is any appreciable overlapping in the period of opening of the inlet valves the first cylinder will always be starved while the second will be surcharged. On the other hand, to avoid overlapping and yet keep the inlet valves open long enough to allow the cylinders to fill up is difficult and necessitates a very late opening indeed. Also the short period introduces difficulty in the case of high-speed engines in regard to operation. Again, such very late opening as is necessary to obtain equality between the cylinders will render the suction of the engine very noisy when running on full load, for this is always an objectionable feature of very late opening inlet valves. It would appear that on the whole the best method of dealing with this very unsatisfactory form of engine is, in the case of comparatively slow-speed engines, to open the inlet valves very late; and in the case of high-speed engines, to employ two altogether separate carburettors and throttles. In either case it is most desirable to fit two separate exhaust pipes, for overlap between the exhaust valves cannot possibly

be avoided, and unless two pipes are fitted, cylinder No. 1 will discharge high pressure and highly heated exhaust products into No. 2 just before the completion of the exhaust stroke, thus filling the clearance space of this cylinder with exhaust gas under pressure, at the one point in the cycle at which the presence of highly heated gas is most undesirable. In some instances a fairly uniform power output may be obtained from such an engine because the starving of No. 1 is balanced by the drowning out with exhaust products of No. 2, with the result that on full throttle both cylinders give a much reduced but more or less equal performance, though in such cases the evils no longer balance one another on reduced loads, when the presence of an excess of exhaust products is more than usually objectionable.

CASE 3. Four cylinders drawing from a single carburettor.

Except that the flow in the branch pipe from the carburettor is relatively constant, this case is almost as difficult to deal with as the last. So far as valve timing is concerned, either all overlap of the inlet valves must be avoided, with the resultant difficulties of an unduly short opening period and noise when running on full throttle, or a certain amount of irregularity, coupled with a reduction in the maximum power output due to robbery of one cylinder by another, must be tolerated. In a four-cylinder engine, however, the branch pipes from the throttle to the several cylinders are generally of considerable length, and in some cases use can be made of the kinetic energy in these branches to fill any one cylinder despite attempted robbery by another, more especially so in the case of very high-speed engines. So far as throttled conditions are concerned, this case may be regarded as one in which the capacity between the throttle and the inlet valves is infinite, and, therefore, in which a late opening inlet valve is definitely desirable. On the whole, it would appear that, for this type of engine, it is best to use very late opening and relatively early closing valves in the case of moderate-speed engines and those which operate normally at a comparatively low load factor, because this will tend to give more economical running at reduced loads; and to use comparatively early opening and late closing valves for engines which are normally run at very high speeds or at a high load factor despite the fact that for maximum power at comparatively low speeds this setting will be inferior to the late opening and earlier closing.

Probably the best results from every point of view can be obtained when two independent induction systems are used, one feeding the inside, and the other the outside pair of cylinders. This arrange-

ment eliminates all question of overlap, and has been applied by the author in the case of several four-cylinder engines designed to give a very high power output and economy.

The once common practice of permitting overlap between the inlet and exhaust valves of the same cylinder is certainly not to be recommended. The arguments for providing overlap in this manner are (1) to make use of the kinetic energy of the gases in the exhaust pipe to scavenge the cylinder and so obtain a greater weight of " live " gas in the cylinder when running at full power ; and

(2) To lengthen the period of valve opening, with a view to reducing the stresses on the valve mechanism at very high speeds.

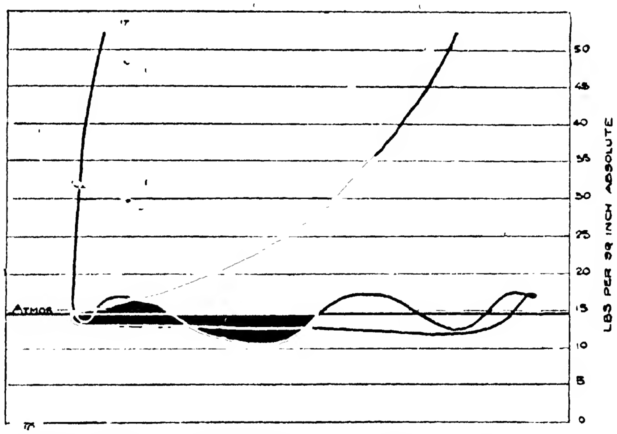

Fig. 74.—Indicator Diagram at 1500 R.P.M. shows Pressure Variations during Exhaust Stroke

The first is wrong in inception and bad in practice, for, in the first place, the gases flow out through the exhaust pipe in a series of pulsations, the pressure ranging usually from about 3 lb. above to 3 lb. below atmosphere, depending upon the length of the pipe, as shown in the indicator diagram, fig. 74, which is taken from one cylinder of an engine running at 1500 R.P.M. It is just as likely that the pressure at the exhaust valve, at the moment when the inlet valve opens, will be above atmospheric as it is that it will be below, in which case a reverse process will occur and exhaust gases will be driven back into the induction pipe. Hence, any advantage which may be gained at one particular speed of rotation will be counteracted by a corresponding but larger loss at other speeds. Again, the valves are seldom far enough apart in the cylinder to permit of any useful scavenging effect being obtained, fresh gas

merely being drawn out of the inlet port and into the exhaust, and so lost. On reduced loads the practice of overlapping is particularly bad, for it must be remembered that at this period the pressure of the residual exhaust products in the exhaust system is round about atmospheric, while the pressure in the induction system may be half an atmosphere or less, with the result that exhaust products are simply sucked back from the exhaust system into the induction system, and that under just those conditions when the presence of exhaust diluent is most detrimental to efficiency. The second argument need not apply if the valve mechanism be properly designed, as will be shown later. In any event, care should always be taken to prevent adjacent cylinders from exhausting into each other. It is usual nowadays to combine together the exhaust ports of the two central cylinders and to connect the exhaust manifold to the two outside and the central pair; this arrangement is fairly satisfactory, but it is better still to use either three or four quite separate exhaust pipes between the valve ports and the manifold, though in practice this is sometimes inconvenient.

CASE 4. Six cylinders drawing from two carburettors, each feeding one group of three.

This case is simple, there is no overlap of the inlet valve periods, and in so far as valve timing is concerned each group may be treated as three single-cylinder engines.

CASE 5. Six cylinders fed from a single carburettor.

In this case (A) the capacity between the throttle and the inlet valve may be regarded as infinite; (B) overlapping of the opening periods of the several inlet valves cannot, under any circumstances, be avoided; (C) unless long separate branch pipes be provided to each cylinder—which is almost impracticable on the grounds of distribution, confusion of pipe-work, &c.—little or no use can be made of the kinetic energy of the gases in the induction pipe owing both to the excessive overlapping and to the constant reversals of direction of flow in the induction manifold. It will therefore be impossible fully to charge any of the cylinders, or, indeed, to attain a final suction pressure in any of them appreciably in excess of the mean pressure in the induction manifold. Since the capacity between the throttle and the inlet valves is so large, it will pay—particularly on light loads—to employ rather late opening inlet valves in order to reduce to the minimum the negative work in the cycle. In such a case the late opening of the valves will cause very little noise at full load because the suction through the carburettor

ment eliminates all question of overlap, and has been applied by the author in the case of several four-cylinder engines designed to give a very high power output and economy.

The once common practice of permitting overlap between the inlet and exhaust valves of the same cylinder is certainly not to be recommended. The arguments for providing overlap in this manner are (1) to make use of the kinetic energy of the gases in the exhaust pipe to scavenge the cylinder and so obtain a greater weight of " live " gas in the cylinder when running at full power ; and

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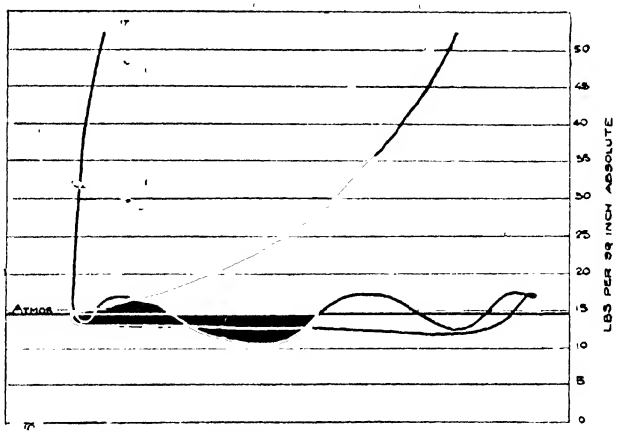

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substantially higher economy obtained thereby. The actual available torque at low speeds is little, if any, reduced, while the torque at high speeds is increased, owing to the longer period of charging.

The cases considered cover practically all the range; where greater numbers of cylinders than six are employed they are always divided into groups, which come under one or other of the categories we have considered.

To sum up, so far as the exhaust valve is concerned, the problem of its design and operation is the same for all engines irrespective of grouping or numbers of cylinders. It should be as small in diameter as possible; the lift should in no case be less than one quarter of the port diameter, and preferably it should be as much as 30 per cent of the port diameter. In all cases it should be lifted and closed as rapidly as possible, while, as regards timing, it is a good rule that it should be at about half lift at the outward centre of the piston and 5 per cent open at the end of the exhaust stroke.

As to the inlet valve, this, too, should be kept small, with a lift not less than one quarter of the port area, in order to obtain the maximum of turbulence. Its time of opening and closing must depend, to some extent, upon the number and grouping of the cylinders, but, except in the case of six cylinders drawing from one carburettor, it should always be closed as rapidly as possible, the primary aim being to keep it nearly wide open at the end of the suction stroke and to close it as soon as possible after. In the case of six cylinders drawing from one carburettor, the inlet valve may be opened and closed much more leisurely.

Cam Design and Valve Operation.— In designing the cams for operating either the inlet or exhaust valves, the primary considerations are both to open and to close the valves as rapidly as possible with the minimum of stress or noise, and at the same time to arrive at a form of cam which can readily and easily be produced. It is only too often that a cam is designed to give a specific opening diagram, and to provide, say, a constant rate of change of acceleration throughout the whole opening period, which may be ideal on the drawing-board but almost impossible to reproduce with accuracy. It must be remembered that a cam profile cannot usually, if ever, be " generated " in the grinding machine, but must be reduced from a master cam, and that the former must be hand-made—the accuracy of its profile can be ensured only when the contour is made up of simple arcs of circles and tangents.

Again, it is always very desirable to avoid any concave surfaces,

since these limit the radius of the grinding wheel which can be used to produce them, thus imposing a very tiresome limit on the manufacture. By a suitable combination of cam and follower the necessity for concave surfaces can always be avoided.

In the operation of a spring-controlled valve by a cam, the first movement of the cam imparts a positive acceleration to the valve until nearly half lift, when the acceleration changes and becomes negative—the valve is then under the control of the spring, whose tension must be sufficient to overcome the inertia due to acceleration. From about half lift to full lift, and from full lift to half closed, the valve is entirely under the control of the spring. For the first half of the lift and the latter half of the closing period the spring is inoperative and the movement of the valve is controlled directly by the cam. The spring, therefore, does not come into effective operation until the valve is nearly half open, and ceases to operate when the valve is about half closed. The rate of acceleration permissible while the valve is under spring control is governed by the pressure and rating of the spring, but during the first and last portions of the valve's movement the rate of acceleration is governed solely by the permissible pressure against the flank of the cam. To make the best use of the spring material the rate of acceleration during the spring-controlled period should be such as to correspond as nearly as possible with the rating of the spring—that is to say, the rate of acceleration should increase steadily as the spring is further compressed. During the first and latter portions of the valve's movement the rate of acceleration may generally be much greater, but should be kept more or less constant.

These considerations indicate that constant acceleration throughout the whole period is by no means ideal; or even desirable. The acceleration during the period of cam control may usually be very high and more or less constant, while the acceleration during spring control should be as low as possible in order to use light springs, and should vary uniformly with the rating of the spring. In any case, of course, the acceleration must be limited to that at which the spring pressure will always overcome the inertia of the valve, and that by a margin sufficient to cover any friction of the valve in its guide.

Fig. 75 shows a convenient form of sheet for setting out cam designs. On this sheet are shown :

1. The acceleration, both positive and negative, during the opening period.

2. The corresponding velocity curve.

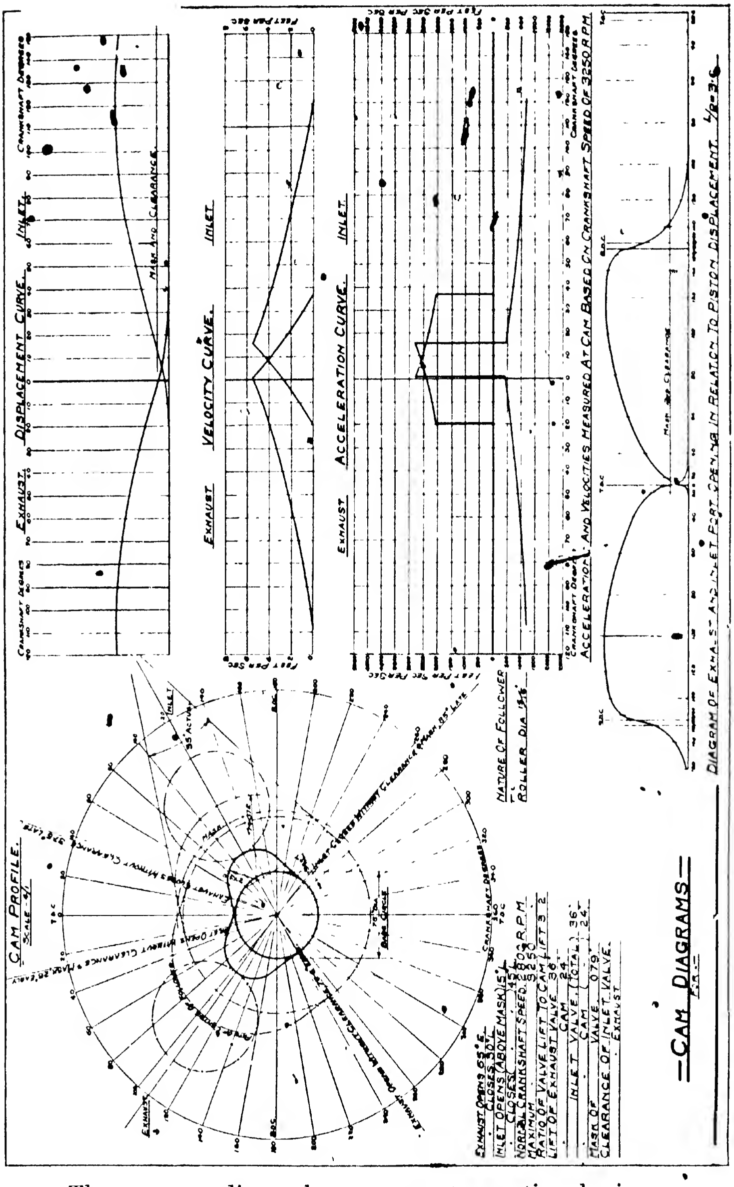


Fig. 75 — Cam Diagrams

3. The corresponding valve movement on a time basis.

4. The valve movement in relation to piston displacement, i.e. the valve opening diagram.

5. The evolution of the contour of the cam.

Permissible Acceleration.—This must be considered from two aspects—the highest permissible acceleration while under spring control and the highest permissible acceleration under direct cam control. Both, of course, depend largely upon the total reciprocating weight of the valve and its gear, which must include half the weight of the spring.

With regard to the acceleration under spring control—this is determined by the weight of active spring material and the permissible stress in the material. In the author's experience, so long as the stress in the material is kept down to from 40,000 to 50,000 lb. per square inch, ordinary spring steel coil springs will last almost indefinitely, even in the highest speed engines. When the valve is small, i.e. less than 2·0 inches diameter, and is operated more or less directly from the cam, and when the weight of intervening gear, such as rockers, tappets, push-rods, etc., is comparatively small, a maximum acceleration of about 1800 ft. per second per second, corresponding to a spring pressure of 56 times the reciprocating weight when the spring is fully compressed, is usually permissible, though except in excessively high-speed engines it is seldom necessary to employ so high a rate of acceleration. The acceleration at the point when the spring first takes up the load must, of course, be lower, in proportion to the rating of the spring. In the case of moderate-speed engines there is no need to employ anything like such a high rate of acceleration when under spring control, and for engines of about 15 to 20 B.H.P. per cylinder running at maximum speeds not exceeding 2000 R.P.M. an acceleration under spring control of 800 to 900 ft. per second per second will permit of as favourable a valve opening as can be desired. At the other end of the scale, a limit is fixed for minimum spring tension; this must always be sufficient, in the case of the exhaust valves at all events, to resist the vacuum formed in the cylinder when running throttled. In practice it is found that in order to prevent the exhaust valves from being sucked open, particularly when the engine is in a state of vibration, a spring tension of at least 11 lb. per square inch is required, reckoned on the area of the head of the valve. In the case, therefore, of a valve the area of whose head is, say, 3 square inches, a minimum spring tension of 33 lb. will be required when the valve is on its seating. With a spring of normal rating, the tension, when the valve is fully lifted, will be at least 50 lb. Now the reciprocating weight of such a valve and its attendant gear will probably be in the neighbourhood of 1·5 lb., and the maximum permissible rate of acceleration in such a

case will therefore be $\frac{50 \times g}{1 \cdot 5} = 1080$ ft. per second per second. This, however, makes no allowance for friction in the guide, but even after making a generous allowance for this factor it will be seen that the case for all moderate-speed engines, when the valves are directly operated and the intervening gear is not heavy, is easily met by the provision of a spring of only just sufficient strength to prevent the exhaust valve from being sucked open when running idle.

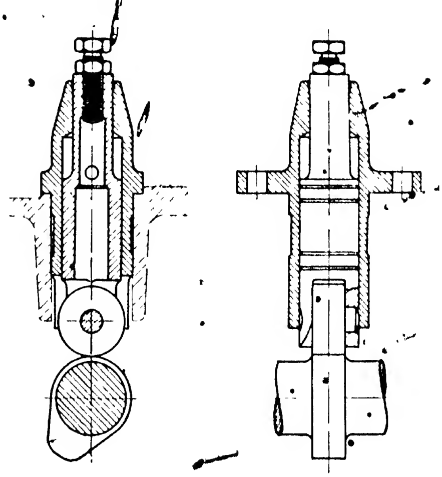


Fig. 76.—Roller Ended Follower

The highest permissible rate of acceleration while the valve is under direct cam control depends upon the type of follower used. This may be either a roller, a curved slipper, or a plain flat-footed or "mushroom"; examples of each of which are shown in figs. 76, 77 and 78. At first sight, it might appear that the roller is the most satisfactory form, but on investigation it will be found that this is far from being the case, and for three reasons:

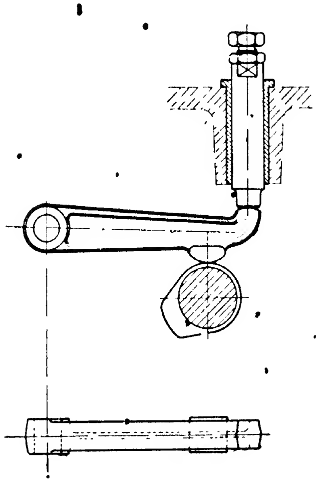


Fig. 77.—Slipper Type Follower

(1) The whole of the load is taken on the roller pin whose projected area is necessarily very small, this pin cannot conveniently be pressure lubricated, and its facilities for obtaining replenishment of oil are very poor, hence it is easily overloaded.

(2) Owing to the changes in surface velocity as the cam revolves and to the inertia of the roller itself, it follows that the latter cannot truly roll, but must skid, and that just at the period when the pressure on it is at a maximum.

(3) The use of a roller greatly increases the weight of the tappet

gear. For moderate-speed engines, when the loads are comparatively light, the use of a roller is permissible, but it should never be used in very high-speed engines for the reasons stated above.

The second type, namely, the curved slipper, is better than the roller, in so far that it involves no bearing which may become overloaded and break down, but it has the disadvantage that it presents only a very small area of rubbing surface against the cam, and so is liable to wear. Both the roller and the slipper "skid," but the latter skids much more rapidly and presents only one face, while the former skids slowly and presents a continual change of face. Against this, however, must be set the fact that both the radius and the width of the slipper can be much greater than that of any roller. On the whole, the roller has the advantage on the score of wear so long as the pressure is light and the pin bearing is not overloaded, while the slipper scores when the rate of acceleration, and therefore the pressure, is high, for though it may wear considerably and therefore require renewal, it will not break down altogether.

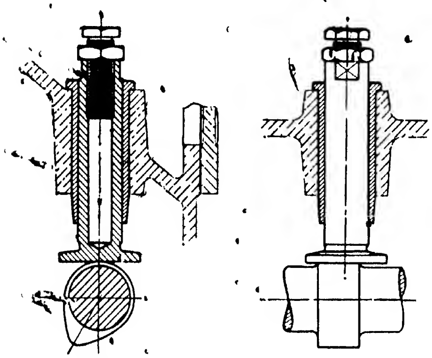


Fig. 78.—Mushroom Type Follower

The third type, namely, the flat-footed or "mushroom" tappet, is, in the author's opinion, the most satisfactory of the three, but it, too, has certain limitations, for it necessitates the use of a cam with a larger base circle, which cannot always be provided. Followers of this type should always be offset sideways, so that the sliding of the cam tends to rotate them. Under these conditions practically the whole surface of the flat foot is made use of, and the wear is less than with either of the other two types, while there is no pin-bearing to be overloaded. It has all the advantages of the roller type in that it constantly presents a new surface in contact with the cam, and all the advantages of the slipper type, in that it has no bearing to fail, while the conditions as regards lubrication are ideal. The one practical objection is that it is usually impossible to employ a low rate of positive acceleration when, if ever, this is desired; hence it is difficult to obtain such quiet running as with the other types, though with careful design the difference is very slight. With flat-footed tappets

it is perfectly safe to employ a very high rate of acceleration, for the pressure occurs only when the flank and not the tip of the cam is in contact with the tappet. It is perfectly safe with the latter type, with cams, say, ⅝ in. wide and a tappet diameter of, say, 1¼ in., to apply an average pressure of 250 lb. during the period of cam control. In the case of a valve and its gear weighing 1·5 lb. with a spring tension at rest of, say, 40 lb., this will correspond to a rate of acceleration of about 4500 ft. per second per second. With the roller type it is very doubtful whether it would be safe to exceed an acceleration of about 2000 ft. per second per second. Since, from the point of view of wear, it is only the average pressure which need be taken into account, it follows that there is no particular advantage to be gained by keeping the acceleration constant during the period of cam control, and that there is no objection to following the line of least resistance and making the flank of the cam either a tangent or a simple circular arc.

It is desirable to keep the base circle of any cam as small as practicable in order to reduce the rubbing velocity between the cam and its follower, and this applies whether the follower be a roller or a slipper.

It must be remembered always that the "effective" radius of any cam is the actual radius from the centre of the camshaft to that of the roller or to the centre of radius of the slipper. Hence it matters not, so far as the valve motion is concerned, whether the cam be large and the roller small, or *vice versa*, except when a flat-footed follower is used. In this latter case it is necessary to use a cam of comparatively large diameter, but when flat-footed followers are used the wearing surfaces are so large and the facilities for lubrication so good that a high rubbing velocity is much less objectionable.

The author is greatly indebted to one of his assistants, Mr. R. J. Cousins, for the construction and development of the following method for arriving readily at the most suitable cam contour, to comply with any given set of conditions.

When setting out the design of any cam the first question is that of deciding whether the cam profile, as dictated by the valve opening requirements, is permissible mechanically, rather than constructing a cam to conform to some ideal figures for positive and negative acceleration. This being the case, an analysis was made of the general form of cam in which the flanks and nose are composed of circular arcs or straight lines, and a series of graphs prepared, giving

practically on sight the actual acceleration at any point for all reasonable proportions of cam.

Fig. 79 shows the acceleration on tangent flanks (dotted curve) and round noses with circular followers, also harmonic cams with flat or mushroom followers.

Fig. 80 gives the figures for hollow flanks with circular followers.

Fig. 81 deals with round (convex) flanks with circular followers.

Fig. 82 shows various forms of cams, internal and external, and indicates the graph to be used in each case.

The formula is the same in all cases, viz.:

$$\text{Acceleration in ft. per sec. per sec.} = \frac{R \times C \times N^2}{100{,}000},$$

where R = radius in inches (see fig. 82),

C = a coefficient from the corresponding graph, depending upon the form of cam and the angle from the base circle (for flanks) or apex (for noses),

N = revs. per minute or crankshaft (assuming camshaft runs at half-engine speed).

A preliminary lay-out of the cam is first made, taking the known factors (which usually include the approximate base circle diameter, lift, period of opening, and room available for roller or follower).

The positive acceleration at the beginning and end of the flank and the negative acceleration (on the springs) at beginning of nose and apex are then read off, and the form of the curve visualized from the suitable graph, when it will be obvious at once if the cam is out of court mechanically, in which case suitable compromises must be made.

Assuming that a tangent cam has been constructed in the first instance, a certain amount of adjustment of the positive acceleration may be made by altering the distance from camshaft centre to the centre of the roller or slipper, the acceleration varying directly in proportion, but if this would make the roller too large on the one hand or make the curvature of the roller or slipper too sharp on the other, the flank may be made curved, concave, or convex as may be necessary, to increase or decrease the positive acceleration.

This will tend to have the opposite effect on the negative acceleration on the nose of the cam, for it will give either more or less time in which to bring the valve parts to rest from maximum velocity.

In cases where the figures are too high all round, it becomes imperative to increase the period to the longest possible and to decrease

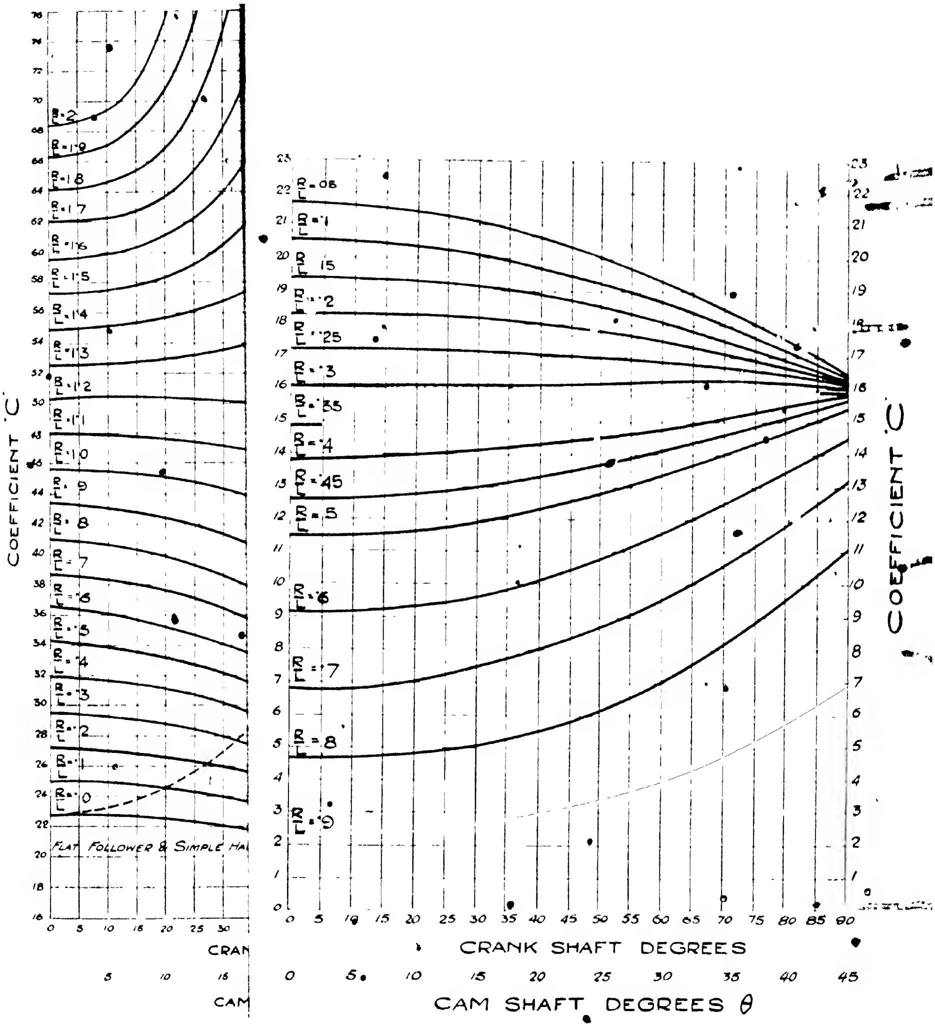


Fig. 81

[Vol. II., facing p. 202.

the lift to the minimum, at the same time providing ample surface on the cam and follower to take the heavy loading.

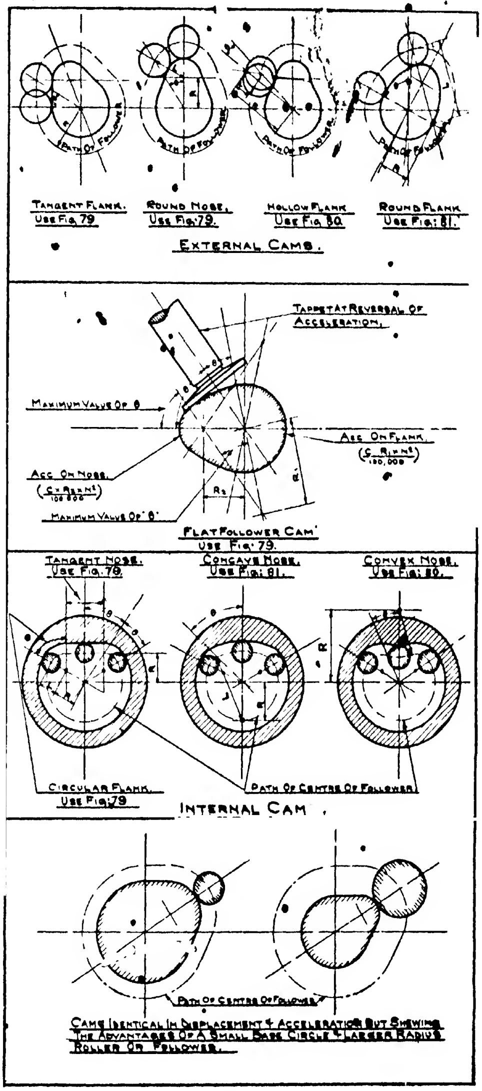


Fig. 82

The negative acceleration on the nose is directly proportional to the distance from camshaft centre to the centre of curvature, it

varies also (but not directly) as the proportion is altered between that distance and the radius of curvature measured to the centre of roller or slipper.

These factors provide a ready means of adjusting the negative acceleration, but it is limited inasmuch as a long radius of curvature cannot be used if the cam period is short or the flank convex.

On the other hand, the radius may be shortened down to a figure very little more than the radius of the follower if the engine speeds and lift are low. This may make the nose of the cam concentric with the shaft for a short distance, giving an improved opening diagram.

It should always be remembered that the nose of the cam does not sustain any pressure at maximum speed because the pressure exerted by the springs should then balance the inertia of the valve parts, while, at low speeds, the pressure on this part of the cam must always be somewhat less than the spring pressure. On the other hand, the flank of the cam has to take the pressure of the spring, the gas pressure on the valve (in the case of the exhaust), and also the force necessary to accelerate the valve parts.

It follows that sharp curvature on the nose of the cam will not lead to undue wear or surface cracks, but that the curvature of the cam flank and follower should be as large as possible.

This consideration points to the use of as small a base circle and as large a roller or slipper as possible for any virtual cam, i.e. for any particular line of motion plotted at the centre of the follower.

As the flank will usually be a straight line or a long radius curve, it is affected little if at all by any reduction in the diameter of the base circle, the nose, as explained, is loaded less and less as the speed increases and need not therefore be considered ($\frac{1}{16}$ in. radius will stand quite well), while the roller or slipper is greatly improved by the increase in its radius of curvature, and the rubbing velocity is also reduced.

In all cases of cams and followers formed of circular arcs the velocity and acceleration of the centre of the follower are the same as those of a piston having a crank and connecting-rod of lengths R and L respectively (see fig. 83). Whenever possible the proportions of the nose of the cam should be such that $\frac{R}{L}$ is less than unity.

This ensures that the negative acceleration will be at a maximum at the apex and fall off towards the points of junction with the flanks. Since all ordinary forms of spring give an increasing pressure towards the top of the lift, a cam nose so proportioned permits

of the use of a spring which approximately balances the inertia at all points.

If, on the other hand, $\frac{R}{L}$ is greater than unity the acceleration increases towards the flanks, and as the spring must be at least equal to the inertia at any point, it follows that it is too strong at the apex and throws an unnecessary stress on the valve gear.

Flat or "Mushroom" Type Followers.—The cams for use with these are most conveniently constructed of circular arcs, one of small radius for the nose and two others of much larger radius placed symmetrically on either side and tangent to the nose and base circles.

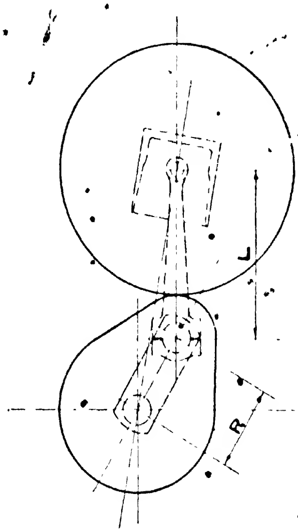


Fig. 83

The arcs of the flanks and nose being continued round to form complete circles will be seen to form cranks or eccentrics with which the flat-ended tappet engages in turn. The motion of the tappet is therefore composed of portions of simple harmonic motions of varying amplitude, and the radial velocities and accelerations about the flank and nose are proportional to the distance from the camshaft centre to the centre of curvature in each case.

This is a useful feature of this form of cam for it enables one to determine a relationship between positive and negative velocities in the first instance and plot the cam accordingly.

A convenient method of construction is appended (fig. 84).

Construction of Harmonic Cams
(see fig. 84)

Cam profile formed of circular arcs.

Follower made with a flat face to engage with cam and moving in a straight line normal to that face.

Draw a horizontal line AO of unit length, say 1 in.

Draw BO and CO forming angle AOB, AOC, so that each equals half the valve period plus clearance. (Clearance may be assumed

to be 12° to 16° total (crankshaft degrees), i.e. 3° to 4° a side actually on the cam.)

On BO produced mark off OD, making OD . OA :: acceleration on flank of cam : acceleration on nose (springs). (This proportion may be determined at first or modified after a preliminary layout. Average cases range from 2 : 1 to 3 : 1.)

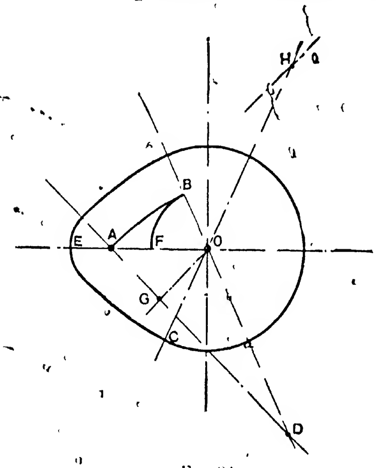


Fig. 84.

With D as centre, DA as radius, describe arc AB.

With O as centre, OB as radius, describe arc BF.

Then as AF : the required lift :: OB : required true base circle radius. (*Note*.—The true base circle radius is smaller than the cam profile by an amount equal to the radius of the nose.)

Construct a similar figure with true base circle and lift as required. Then take a suitable radius (say $\frac{1}{16}$ in. or more) and describe an arc with centre A forming the nose of the cam, and complete the profile by arcs from centres D, O, and H.

The size of the tappet head is found as follows:—

Join AD and draw OG perpendicular to AD.

Then OG is the maximum eccentricity of the point of contact, and occurs when the tappet is on the junction of the flank and nose curves (i.e. the point of reversal of acceleration from positive to negative).

The radius of the tappet head must be slightly greater than OG, so that the cam may not overrun the edge.

The changes may be rung very easily on the relationship of flank and nose because with this type of cam and follower very heavy positive pressures may safely be used without producing localized wear on the tappet. It will be found that high positive accelerations and long opening periods produce small base circle cams.

Every effort should be made to keep the base circle small, and to this end the radius of the nose may be reduced to $\frac{1}{16}$ in. if necessary with safety. Another convenient property of these simple harmonic cams is that the negative acceleration is greatest at the apex and falls off in a regular manner towards the point of junction with the

flank, so that if the spring pressure at the apex be made to balance the inertia at that point and the total deflection from free length to maximum compression be equal to the distance from camshaft centre to centre of nose, the spring pressure will exactly balance the inertia at all other points. This renders the determination of the most suitable spring a very simple matter.

Internal Cams.—These had a considerable vogue at one time on small single-cylinder engines. When composed of circular arcs and straight lines they follow the same laws as external cams of the same $\frac{R}{L}$ proportions, but it should be noted that the actual cam is larger than the virtual cam or line of motion of the roller centre, whereas the ordinary external cam is smaller, so that for any particular case the internal cam has a much higher rubbing velocity.

Moreover, since the cam must embrace the roller or slipper the latter is necessarily very much limited in size and the rubbing speed on the pin considerable when a roller is used.

By the use of very good material and workmanship, and particularly by their success in producing excellent surfaces on the pins and rollers, some manufacturers have succeeded in obtaining very satisfactory results, but the type is certainly not to be recommended from either the theoretical or the manufacturing point of view, the internal grinding alone being sufficient to give the decision in favour of external cams.

The graphs given in figs. 79, 80, and 81, and the key diagrams in fig. 82, are worked out for all reasonable proportions, but the formulæ are given below so that extreme cases may be dealt with. True radial movement of the tappet is assumed in all cases. Where the tappet takes the form of a lever, the fulcrum must be so placed that the arc followed by the roller centre approximates to a radial line, otherwise serious distortion of the valve opening diagram will occur, and stronger springs will be necessary, owing to the fact that the acceleration is increased on one side of the nose and decreased on the other, as compared with the value for a radially moving follower.

The acceleration in all cases corresponds to the second differentiation of the radial displacement in respect to time and for the regular forms here dealt with is as follows:

Straight line tangent to base circle

$$\text{Acc.} = W^2R\,\frac{(1 + 2\tan^2\theta)}{\cos\theta} \quad \ldots \quad \ldots \quad \ldots \quad (1)$$

Round nose, and round or hollow flank

$$\text{Acc.} = W^2R\,\frac{(\cos\theta + n^2\cos 2\theta + \sin^4\theta)}{(n^2 - \sin^2\theta)^{3/2}} \quad \ldots \quad \ldots \quad (2)$$

Simple harmonic cam with flat follower

$$\text{Acc.} = W^2R\cos\theta, \quad \ldots \quad \ldots \quad \ldots \quad (3)$$

where R = radius in ft. from shaft centre to roller centre when on base circle in the case of (1) and the radius from shaft-centre to centre of curvature in case of (2) and (3),

W = angular velocity in radians per sec.,

θ = angle moved through from point of contact with base circle for flank (1), (2), and (3) and, in the case of the nose; the angle from the apex = 180° − θ (2) and (3),

L = radius of curvature,

$n = \frac{L}{R}$ (see accompanying figures).

Masked Valves.—It will be evident that as the period required for the inlet valve opening is shorter than that for the exhaust, the accelerations will be greater, increasing inversely as the square of the time. It will also be noted from any ordinary valve diagram that as the velocity of the valve is zero at beginning and end of its period the value of the opening is very small for the first and last 20 or 30 degrees. By recessing the valve seat in such a way that the outer diameter of the valve acts as a piston valve, it is possible to start the motion earlier and finish it later. While keeping the time of opening and closing as before (because the valve head must clear the recess before any appreciable quantity of gas can pass) this greatly reduces the acceleration and usually permits of the use of the same cam for inlet and exhaust. It has also a considerable effect on the valve opening diagram since the end of the diagram, instead of being attenuated, retains a considerable value and finishes abruptly.

From the point of view of volumetric efficiency this is the most useful feature of the recessed or "masked" valve, as it is usually called, because it gives a large opening at bottom dead centre and closes the valve before the piston has risen sufficiently to pump the charge back through the valve. In normal cases a mask depth of about ¼th to ⅛th of the lift is suitable.

Valve Springs.—These must be considered from four points:

(1) The force at various points in the travel.
(2) The maximum stress in the wire.
(3) The stress range from max. to min.
(4) Periodic vibrations in the spring itself.

(1) From the displacement and acceleration diagrams a force/lift graph is drawn as follows:

A number of points are taken on the displacement diagram from apex to point of reversal (where the nose joins the flank) and are

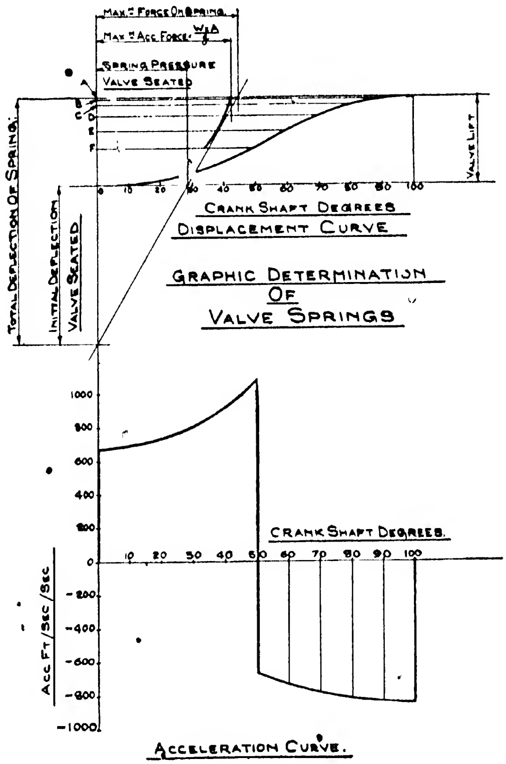


Fig. 85

projected down to the acceleration diagram and also horizontally to a vertical line.

On the horizontal lines are marked off the distances A, B, C, D, &c., equal to the corresponding vertical ordinates on the acceleration diagram. These points are joined by a curve which shows the force necessary to balance the negative acceleration in terms of the lift (see fig. 85).

Practically all valve springs follow a straight line law, and therefore the straight line which most nearly corresponds to the force/lift graph represents the most suitable spring.

If A is the acceleration at any point in ft. per second per second the force should be $F = \frac{A \times W}{32 \cdot 2}$ plus a small allowance for friction. W is the weight in lb. of the valve, tappet, etc., including half the weight of the spring. (This latter figure may be assumed and a subsequent correction made if necessary.) The minimum force must be sufficient to hold the exhaust valve shut when the throttle is almost closed and to avoid unnecessary loading on the valve gear, the maximum should be just high enough to allow a safe margin beyond the highest engine speed.

(2) The maximum stress in the wire must be well below the yield point, otherwise permanent set will take place and the free length will become less, thus reducing the force required to compress the spring to any given point.

It is not advisable to exceed 30 tons per square inch, and it is always preferable to keep within 20 to 25 tons per square inch maximum.

(3) The stress range, i.e. the difference between the initial stress (valve closed) and the maximum stress (valve fully open), should be kept down in order to avoid fatigue and rapid deterioration of the metal of the spring. The range should not exceed 12 to 15 tons per square inch according to the quality of the steel and the life expected.

(4) If the mass of the spring itself be too great in relation to its stiffness (which is proportionate to its rate, i.e. the force in lbs. required to compress it 1 in. axially) the natural period of oscillation of the spring becomes large and may even approach the period of the valve motion. Serious vibrations may then be set up, and the spring is liable to fatigue and may allow the valve gear to jump the cam.

The number of free vibrations per minute of the centre of the spring when the ends are held is:

$$N = 513\sqrt{\frac{R}{W}},$$

where N = number of vibrations per minute,

R = rate of spring, i.e. lbs., required to compress it 1 in. axially,

and W = the weight of the spring in lbs.

If it is found that N is equal to the revolutions per min. of the camshaft or is a simple multiple of same (say 2, 3, or 4 times), it is

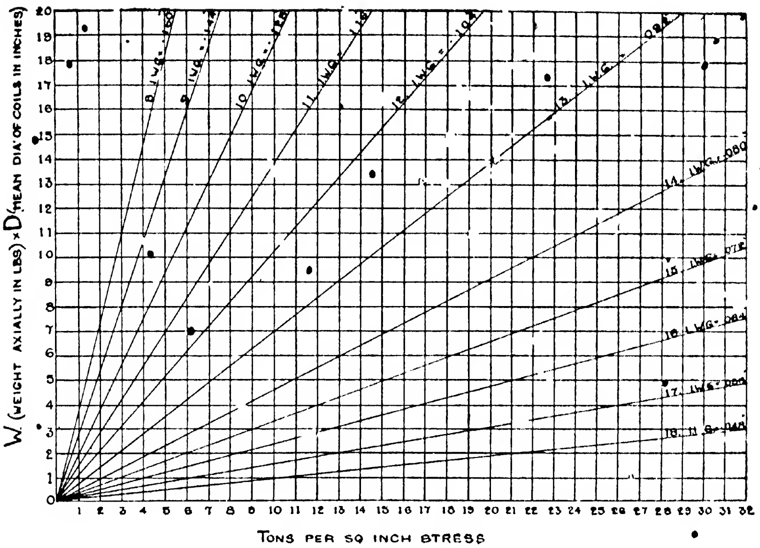


Fig. 86.—Spring Graph—Stresses

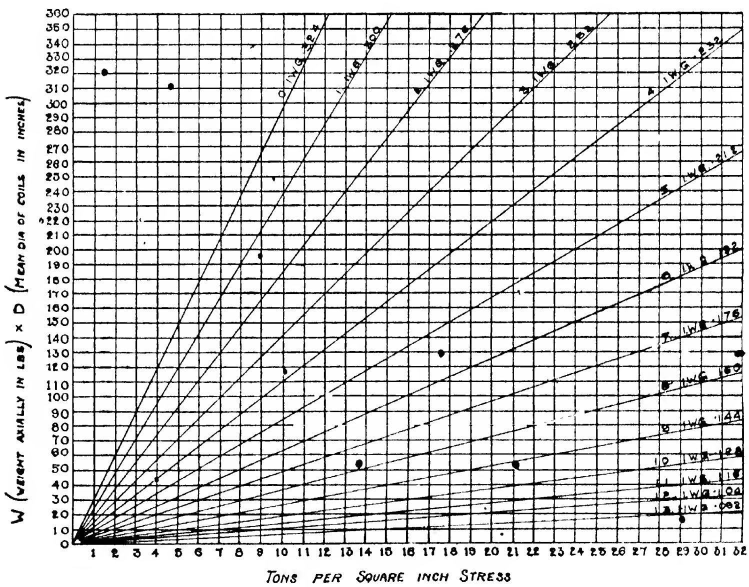


Fig. 87.—Spring Graph—Stresses

practically certain that the spring will shudder and cease to function properly.

A vibrating spring appears blurred when the engine is running, whereas in a spring which is functioning correctly, the central coils can be seen clearly owing to the fact that they are stationary while the valve is shut (say, two-thirds of the total time) and the eye retains the impression.

The accompanying graphs will assist in the selection of the gauge of wire (see stress graph fig. 86 for light gauges and fig. 87 for heavier gauges) when the force and approximate diameter of spring are

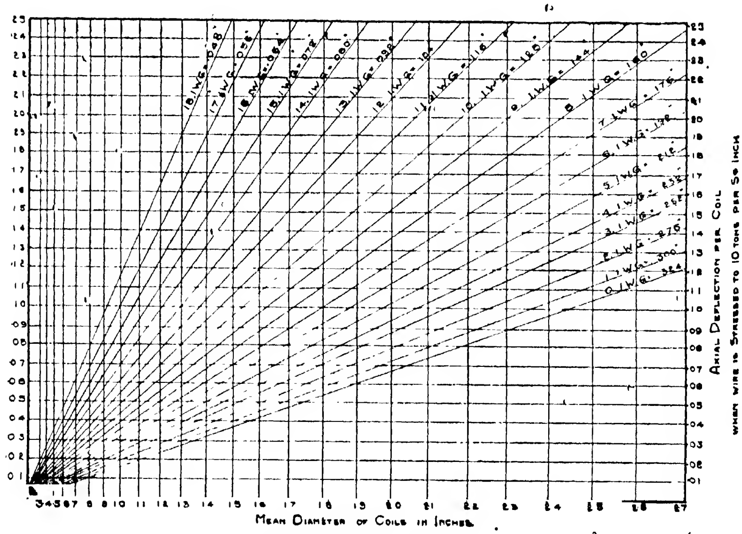


Fig. 88.—Spring Graph—Deflection

known. The deflection per coil (d) may be read off from the deflection graph (fig. 88). The total deflection (D) being already fixed (see figure for force/lift graph) the number of *effective* coils is $\frac{D}{d}$ and the total number $\frac{D}{d}+2$.

$$\text{The maximum stress} = \frac{8FD}{\pi d^3} \text{ lb. sq. inch,}$$

where F = maximum force in lbs.,

D = mean dia. of coils in inches,

d = dia. of wire in inches;

$$\text{also the maximum deflection} = \frac{8FND^3}{Cd^4},$$

where N = effective number of coils (total minus 2),

and C = the transverse modulus of elasticity = 13,000,000.

The spring thus arrived at should then be considered under heading (4) before being passed as suitable.

It will be apparent that (3) and (4) are in opposition, for if the stress range be kept *too* low, the spring loses stiffness and may develop slow vibrations in time with the valve gear. There is no difficulty in practice in satisfying both points even in very high-speed engines.

CHAPTER IX

PISTON DESIGN

In the first volume of this book the question of piston design was dealt with at considerable length, and though written some eight years earlier, the principles laid down and the views expressed therein hold good, generally speaking, at the present day. Broadly speaking, the main objects to aim at in the design of a piston for the lighter high-speed types of internal-combustion engine are :—

(1) To reduce friction to the lowest possible limit.

(2) To reduce the weight.

(3) To dissipate heat to the walls of the cylinder.

(4) To prevent the passage of oil into the combustion chamber.

(5) To provide adequate support for the gudgeon pin.

As has been shown in the first volume, conditions 1 and 2 are largely interdependent, for the bulk of the average pressure exerted by the piston against the cylinder walls is, in any high-speed engine, due to the resolved component of the inertia forces which, when averaged over the whole cycle, exceeds the fluid pressure; hence if the weight is reduced the average bearing pressure is reduced also, and for the same bearing pressure per square inch, and therefore for the same durability, the area of bearing surface may be reduced nearly in proportion to the weight.

Piston friction is of course dependent also upon the nature and condition of the lubricating oil adhering to the cylinder walls.

The general question of lubrication and friction has been considered in relation to bearings, etc., in Chapter V, and it has been shown that friction is, to a large extent proportional to the area of surface, the viscosity of the lubricant, and, to a much less extent, to the load. In the case of the piston, however, the conditions are somewhat different; in the first place, although the rubbing velocity is higher, the average load is low and is, compared with any of the bearings, small. Under such conditions the area of surface and the viscosity of the lubricant play a very important part.

With regard to the area of surface, it is clear that only the surface at right angles to the line of the crankshaft is operative, the surface at the sides of the piston receiving no thrust at all. It is therefore clearly desirable to remove all inoperative surface in order to reduce, as far as possible, the area of the oil film in shear. In view of the very light loading to which a piston is subjected, a relatively small area of bearing surface suffices, and for a reasonably light piston an actual bearing surface on either side equal to 50 per cent of the area of the crown should be ample, provided it is properly disposed; that is to say, provided it is disposed equally above and below the gudgeon pin and over a subtended angle of from 90° to 110°. The author has never heard of any case of a piston seizing from overloading of the bearing surfaces. Seventy per cent of piston seizures are due to insufficient allowance for expansion or to distortion, and the remaining 30 per cent to complete failures of the oil supply. As to wear, in the case of cast-iron pistons at any rate, this is mainly due to the piston rings; it is very unusual to find any serious wear on the bearing surfaces of a piston or in the cylinder bore below the travel of the piston rings. In the case of aluminium alloy pistons the position is somewhat different, because the softer metal of the piston permits of particles of grit embedding in its surface, and so grinding or lapping the cylinder walls; also there is evidence to show that some at least of these alloys are liable to form a highly abrasive surface due to segregation of a hard component of the alloy.

It has already been shown in Volume I that, when compared with any other bearing surface in the engine, the friction of the piston is abnormally high; this is undoubtedly due primarily to the fact that the oil is partially carbonized and its viscosity, and therefore its resistance to shear, is greatly increased. It must be remembered that, at every cycle, most of the oil clinging to the walls of the cylinder barrel is exposed to the full flame temperature of the burning gases. It is probable also that the fluid resistance is greater when the direction of motion is constantly reversed than when it is continuous, as in the case of a shaft running in bearings. As an illustration of the effect of the carbonizing of the oil on piston friction, the author has always observed, when testing engines on electric dynamometers, that if the supply of fuel be suddenly cut off after running under load and the engine motored round by means of the dynamometer the friction torque is at first high, but falls rapidly as the carbonized oil on the cylinder walls is replaced with fresh clean

oil from the lubricating system. Fig. 89 shows a typical curve of total friction torque on a time basis carried out under these conditions. The engine in this instance was run under full load at 1200 R.P.M. for a considerable period, until all temperature conditions had become normal; the supplies of circulating water and fuel were then simultaneously cut off and the engine motored round at precisely the same speed—the change over from full load running to motoring being effected without any measurable interval of time and without any appreciable change in speed. In this particular

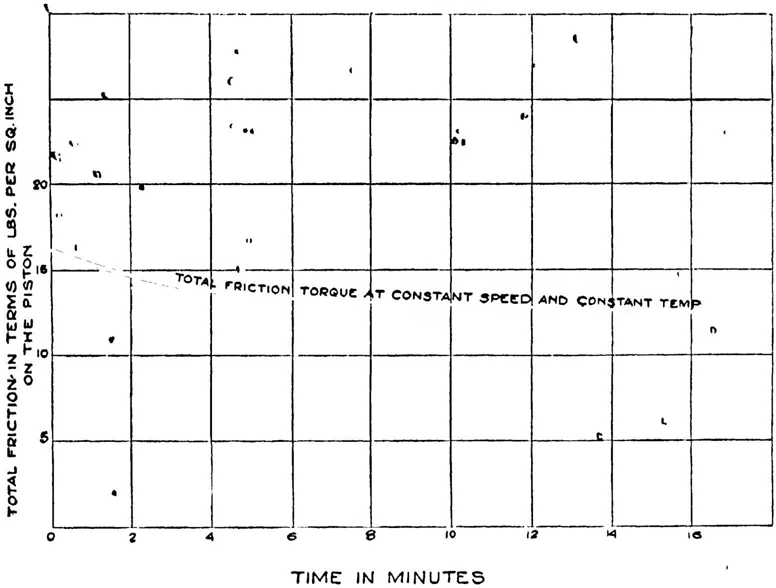


Fig. 89.—Curve showing Drop in Friction Torque as contaminated Oil is replaced by fresh Oil

case the friction losses of the bearings and auxiliaries and the fluid pumping losses had all been ascertained separately, and were found to be equivalent to a mean pressure of 6·5 lb. per square inch at 1200 R.P.M. Deducting these, the piston friction is as shown in the curve, fig. 90, from which it will be observed that it falls from the equivalent of 9·5 lb. per square inch immediately after the fuel is cut off to 6·5 lb. per square inch after ten minutes, by which time it may be presumed that practically the whole of the carbonized oil on the cylinder walls has been replaced by fresh oil.

Influence of Temperature on Piston Friction.—As might be expected, the friction of the piston is largely dependent upon the

temperature of the lubricant, and since the temperature of the latter is determined primarily by that of the cylinder walls to which it clings, it follows that the friction is controlled very largely by the temperature of the cooling water. In Chapter III, when dealing with the influence of cylinder temperature upon power and economy, it was shown that the indicated horse-power of an engine decreases with increase of temperature, because the reduction in the weight of charge far more than outweighs the slight gain due to reduced heat losses. In practice the brake-horse-power and economy of an engine generally increase with increase of temperature, because the

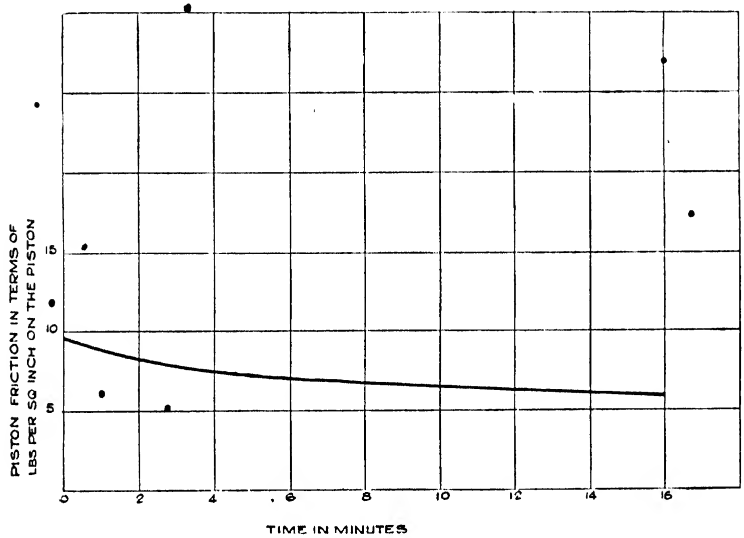


Fig. 90.—Curve showing Drop in Piston Friction alone from fig. 89

reduced piston friction more than compensates for the reduction in indicated power. Fig. 91 shows the results of a test on a standard four-cylinder commercial vehicle engine when motored at a speed of 900 R.P.M. In this test, after the engine had been run for some time the water-jackets were thoroughly flushed through with cold water until the cylinders had been cooled down to atmospheric temperature. The supply of circulating water was then cut off and the rise in temperature and the friction torque (expressed in terms of mean pressure on the engine) were recorded at intervals of two minutes.

In the case of this engine, the friction of the bearings and auxiliaries and the fluid pumping losses were determined separately,

and were found to amount to the equivalent of 5·5 lb. per square inch at 120° F. If this figure be deducted throughout, it will be observed that the piston friction falls from 10·5 lb. per square inch with a cylinder temperature of 70° F. to 5·6 lb. per square inch with a temperature of 150° F. Over this range of temperature the decrease in indicated mean pressure, in the case of this particular engine, is about 2 per cent or 2 lb. per square inch, but the drop in piston friction is equivalent to about 5 lb. per square inch, so that the net increase in power at the higher temperature would be about 3 lb.

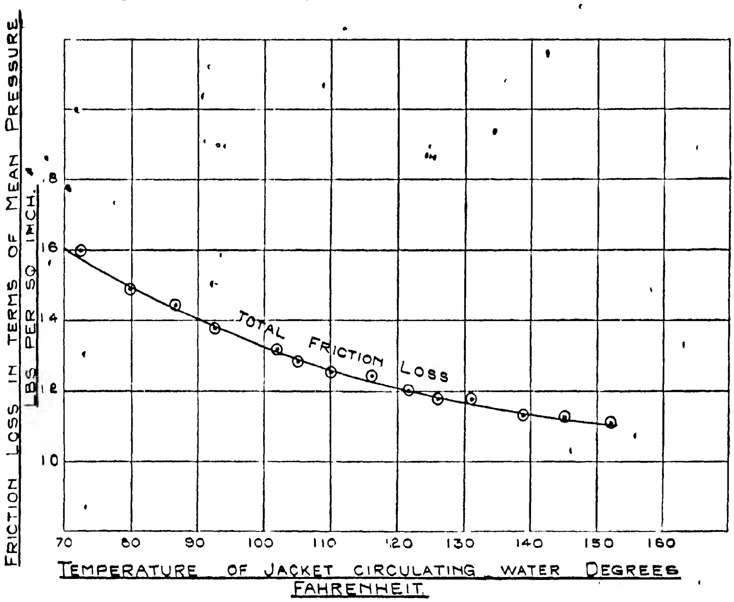


Fig 91.—Motoring Tests, showing Change of Friction with Temperature at 900 R.P.M.

per square inch or 3 per cent. This agreed fairly closely with actual results obtained when running under power, when the difference was found to be nearly 4 per cent. The conditions are not, however, exactly comparable, because—

(1) When running under power the temperature of the piston and inner surface of the cylinder walls is higher than that of the jacket water; this would tend to reduce the difference in piston friction between hot and cold, because the whole temperature scale is virtually raised.

(2) When motoring, the oil on the cylinder walls was clean, consequently the piston friction was lower; under actual running

conditions the piston friction was no doubt about 30 per cent greater at all temperatures; this would accentuate the difference, and probably more than outweighs (1).

Dissipation of Heat.—The heat from the crown of the piston is disposed of—

(1) Through the rings to the cylinder walls

(2) Through the bearing surface to the cylinder walls.

(3) To the oil and air below the piston.

There is a great deal of evidence in support of the theory that, in comparatively small engines at all events, the bulk of the heat passes to the cylinder walls *via* the piston rings. The author could cite numerous experiments in confirmation of this theory, but it is probably sufficient to state that experiments have shown that, when all transference of heat by way of the bearing surfaces has been cut off, the temperature of the piston crown is found to be very little higher. In any case, it is evident that heat can only be transmitted rapidly to the cylinder walls through that portion of the skirt or bearing surface which is being pressed hard against them by the thrust of the connecting-rod and where the oil film is therefore both thinnest and in most active movement.

The proportion of heat carried away by the circulation of the air and oil below the piston cannot be very large and need not be taken seriously into consideration, except in cases where special arrangements are made to increase these effects.

It is quite clear that the most important consideration is the transmission of heat from the centre of the crown to the circumference; once the heat can be conveyed to the circumference, experience shows that there is no difficulty in getting rid of it. In order to facilitate the transmission from the centre to the circumference, it is obvious that the crown should be made as thick as possible consistent with the weight limitation, and the conductivity of the material should be as high as possible. During the last few years the use of aluminium alloys has come into vogue for pistons; not only is their weight about one-third that of cast iron, but their conductivity is about five times as great. With such alloys, it is found that the rate of heat transmission is so high that, even in the case of cylinders developing over 120 B.H.P., there is no need to make the crown of the piston any thicker than is needed for strength alone. Recently all aero-engines, and many others also, have been fitted with pistons made throughout of aluminium alloys. The objections to an all-aluminium piston are :—

(1) That owing to the very high rate of expansion with temperature a large clearance must be allowed; this causes an audible knock when the thrust is reversed under pressure at the end of the compression stroke.

(2) Aluminium being a relatively soft material permits of particles of grit embedding in it, and is therefore liable to cause wear of the cylinder walls unless the latter have a very hard surface.

(3) Aluminium castings, unless very carefully annealed, are liable both to grow and to distort, so that yet more clearance must be allowed on this account.

None of these objections apply when the crown and ring-carrying portion alone is made of aluminium alloy and the bearing surfaces of cast-iron, while advantage can still be taken of the high conductivity and light weight of aluminium.

Passage of Oil into the Combustion Chamber.—One common form of trouble with internal-combustion engines, and more particularly with those of the high-speed enclosed type, is the passage of oil into the combustion chamber, where it carbonizes both on the walls of the chamber and on the crown of the piston, and so gives rise to detonation and ultimately to pre-ignition. Passage of oil past the piston rings and so into the combustion chamber is due to—

(1) The oil is forced up against the rings on the downward stroke of the piston because the motion of the piston, combined with its thrust against the cylinder walls, sets up a very considerable hydraulic pressure and the oil is, so to speak, rolled up against the rings.

(2) The motion of the piston rings in their grooves tends to pump the oil into the combustion chamber.

In order, as far as possible, to prevent the passage of oil into the combustion chamber, the following considerations should be taken into account:—

(1) The setting up of a heavy hydraulic pressure can largely be prevented by perforating the bearing surface so that the pressure can relieve itself, and also by freely venting the piston just below the bottom ring.

(2) As the piston travels downwards the rings are all bearing against the top faces of the grooves, and the clearance between the lower sides of the rings and their grooves is filled with oil scraped from the cylinder walls. As the piston rises again, the rings change over and bear against the lower face of the groove; the oil therefore passes round behind the rings to their upper face and, at the top of the

stroke, when the rings again change sides, some of it is squeezed out. It will be seen therefore that each ring functions as a valveless oil pump and tends to deliver oil into the combustion chamber.

In order to reduce this pumping as far as possible:—

(1) The rings should be made as close a fit as possible in their grooves.

(2) Ample venting should be provided below the lowest ring to permit of the free escape of any oil scraped off the cylinder walls.

(3) The tendency of the rings to pump oil can further be checked to a large extent by drilling holes through the ring groove behind the ring, thus permitting of the free escape of any oil as it passes the back of the ring.

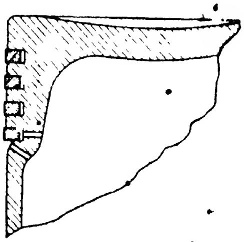


Fig. 92.—Section of Slipper Piston, with Scraper Ring vented by Holes drilled through Groove

This latter expedient should apply only in the case of the lowest ring, since such drilling permits also of the escape of gas. Fig. 92 shows an arrangement of rings, &c., which has been found very effective in preventing the passage of oil into the combustion space.

There is a widespread belief that the passage of oil into the combustion chamber is dependent primarily upon the pressure or vacuum in the cylinder, and that, when throttled down, the reduced pressure in the cylinder during the suction stroke causes oil to be sucked up past the rings. This belief is founded upon the fact that when an engine is opened out after running throttled up, smoke becomes apparent, indicating an excess of oil in the combustion chamber. Also when an engine has been run with the throttle nearly closed and the cylinders or valve caps are removed, oil in a liquid state is then found in the combustion chamber. In spite of such evidence, however, the theory is quite fallacious, for the actual quantity of oil passing the rings is found to be a function of the speed, and of the speed alone, the pumping pressure set up by the rings being of far too high an order to be influenced appreciably by any relatively slight differences of pressure in the cylinder. When an engine is running at or near its full load, the oil passing into the combustion chamber is burnt along with the fuel; combustion is nearly complete, so that no smoke is visible from the exhaust. When running dead light the flame temperature, owing to the large dilution with exhaust gases and the relatively higher rate of heat

loss, is insufficient to burn the oil, with the result that it accumulates in the combustion chamber until the throttle is opened; the quantity is then so large that there is not sufficient oxygen available for its complete combustion at first, with the result that it is only partially burnt and issues from the exhaust as blue smoke—that is, as partially vaporized but unburnt oil.

In any normal high-speed closed-type engine, about 90 per cent of the lubricating oil consumed is burnt in the cylinder as fuel, a fact which should always be borne in mind when reckoning the efficiency of an engine, for the hourly consumption both of fuel and oil should be taken into account. In most normal engines, the proportion of oil consumed is very small in relation to the fuel and does not materially affect the consumption of the latter, but in the case of certain aero-engines, particularly of the rotating cylinder type, the consumption of oil is so high as materially to reduce the fuel consumption, and so give rise to a fictitious fuel economy.

The author has carried out a number of tests in order to ascertain the influence of both pressure and speed on the passage of oil into the combustion chamber, and has tried the effect of motoring an engine and collecting the oil passing the piston under the following conditions:—

(1) When the pressure on either side of the piston is atmospheric.

(2) With a continuous vacuum of 20 in. of mercury in the cylinder.

(3) With a continuous air pressure of 45 lb. per square inch above the piston.

In all three cases the quantity of oil passed per hour was, within the limits of observation, the same—certainly to within 10 per cent. In all cases also the quantity of oil passed by the piston varied almost directly as the speed of rotation.

In fig. 93 is shown a special machine used for testing pistons and rings for—

(1) Friction.

(2) Leakage.

(3) Passage of oil.

It consists of a water-jacketed cylinder mounted on a crankcase in which different types of pistons, rings, &c., can be fitted. The piston is reciprocated by means of a crank and connecting-rod, and, to ensure freedom from vibration, reciprocating balance weights, operated by eccentrics, are provided. The cylinder is heat-insulated from the crankcase and the friction of the piston is measured directly by the temperature rise of the water in the jacket. In order to

ensure uniformity of temperature the water in the jacket is kept circulating by means of a small propeller driven by a belt from the crankshaft. The top end of the cylinder is connected to a large and heavily lagged receiver of sufficient capacity to prevent any

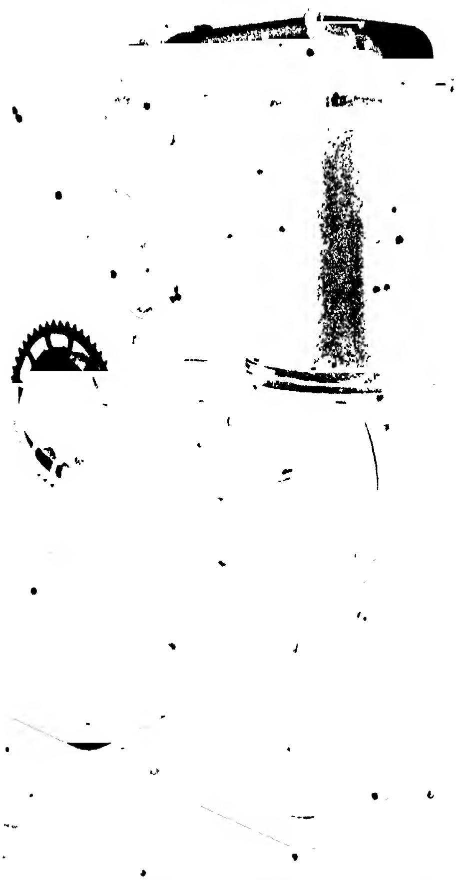


Fig. 93.—Photo of Machine designed for testing Pistons

appreciable variation in pressure, so that the same air is drawn in and out of the cylinder at every stroke, and errors, due to the circulation of cold air inside the cylinder, are eliminated, or nearly so. This receiver is, in turn, connected to an air pump so that the pressure on the piston can be raised or lowered to any desired degree, and the effect of fluid pressure both on piston friction and the passage of oil can be observed.

For lubrication, oil is forced under a pressure of 30 lb. per square inch through the hollow crankshaft, from which it passes out through the connecting-rod big-end bearing and is thrown on to the cylinder walls.

For testing gas tightness and leakage of rings the receiver is removed and a plain cover fitted in its place on the top of the cylinder. This cover is provided with a small and very light automatic inlet valve connected to an air-measuring device. With this cover fitted the piston alternately compresses and expands the air in the cylinder,

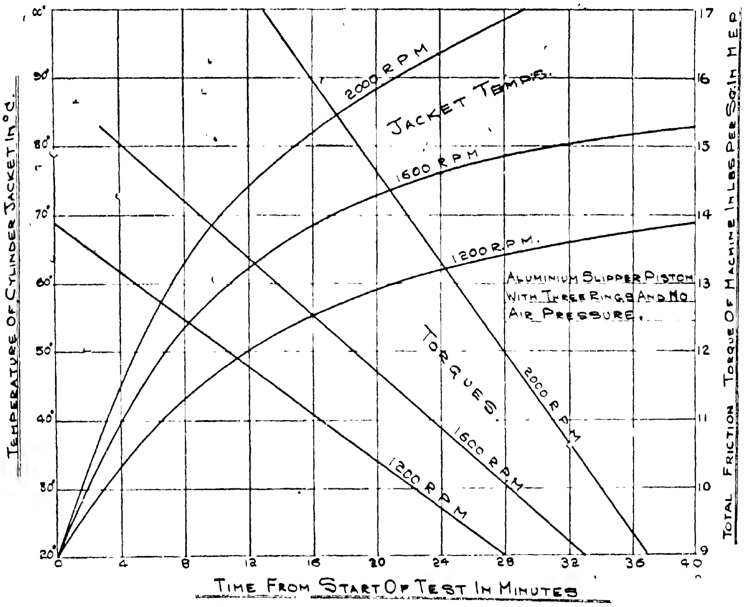


Fig. 94.—Test Readings at three different Speeds

the maximum pressure being about 220 lb. per square inch. Any leakage past the rings is made up by air entering through the inlet valve, and the quantity of air so required to make up for leakage is measured by the displacement of water.

The machine is direct connected to a balanced swinging field electric dynamometer, and can be driven at any speed from 600 to 2500 R.P.M. The total torque required to rotate the machine at any speed can be measured direct from the arm of the dynamometer, while the piston friction alone can be determined directly from the temperature rise of the water in the cylinder jacket.

Owing to losses by radiation, &c., it is not easy to determine accurately the absolute friction of the piston, but the relative friction as between two pistons, or any variations in piston design or rings, can be measured with extreme nicety by comparing the curves of temperature rise of the jacket water. Figs. 94, 95, and 96 show a number of such curves of temperature and time and also the total friction of the machine as a whole in terms of lb. per square inch on the piston, with various types of piston, numbers of rings, &c.

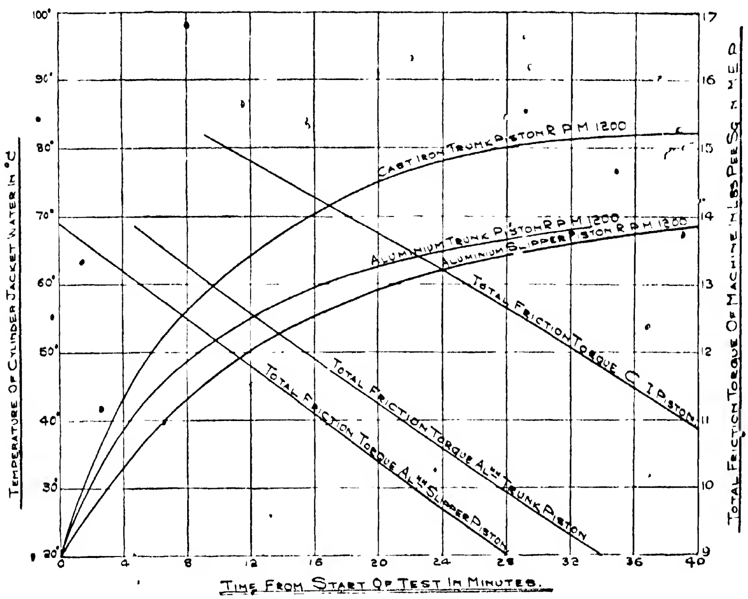


Fig. 95.—Test Readings with 3 different Pistons. Speed 1200 R.P.M. in all cases

In order both to reduce weight and to prevent distortion, it is clearly very desirable to transmit the pressure as directly as possible from the crown of the piston to the connecting-rod, and from the connecting-rod to the bearing surfaces. The customary method of transmitting the pressure from the crown through the side walls and ring grooves to the two extreme ends of the gudgeon pin has nothing to recommend it. It is clearly far better to transmit it directly from the crown of the piston to the gudgeon pin at points as near the centre as the width of the connecting-rod small end bearing will permit.

Fig. 97 shows a design of all-aluminium piston, and represents probably the lightest possible construction. In this design two main ribs transmit the load from the crown to the gudgeon pin, and from the gudgeon pin to the bearing surfaces. Also all unnecessary bearing surface has been eliminated. This type of piston has come to be known as the slipper type, and has been widely used, particularly in very high-speed engines, where its light weight, low friction loss, and effective oil-resisting properties have rendered

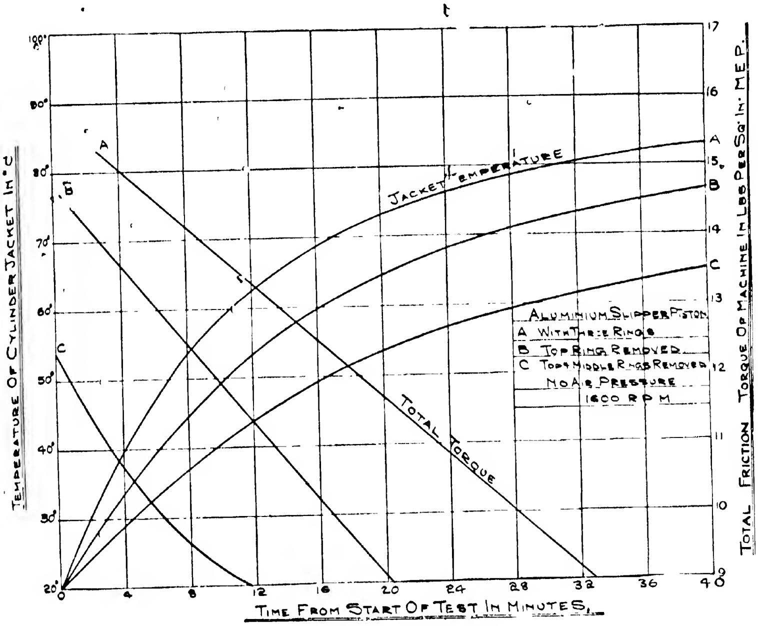


Fig. 96.—Test Readings with different Numbers of Piston Rings. Speed 1600 R.P.M. in all cases

it of great advantage. The example shown in fig. 97 is the largest yet made, for this single piston transmits 135 B.H.P. at 1400 R.P.M.

Fig. 98 shows an alternative design in which the slipper or bearing surfaces are of cast iron. This design has the further advantage in that the floating gudgeon pin is located by the sides of the cast-iron sleeve and requires no other means of endwise location—always rather a troublesome problem.

Fig. 99 shows yet another design, in which the aluminium alloy head is connected with the cast-iron cross-head portion by means of

the gudgeon pin, or rather by means of the bushes in which the gudgeon pin floats. This design has the advantage that it is cheaper and less fragile than that shown in fig. 98; also that the aluminium head is free to centre itself in the cylinder. The chief objection to it is that it necessitates very accurate workmanship.

Piston Knock.—Owing to the large clearance which must be allowed when all-aluminium pistons are used, it is very difficult to

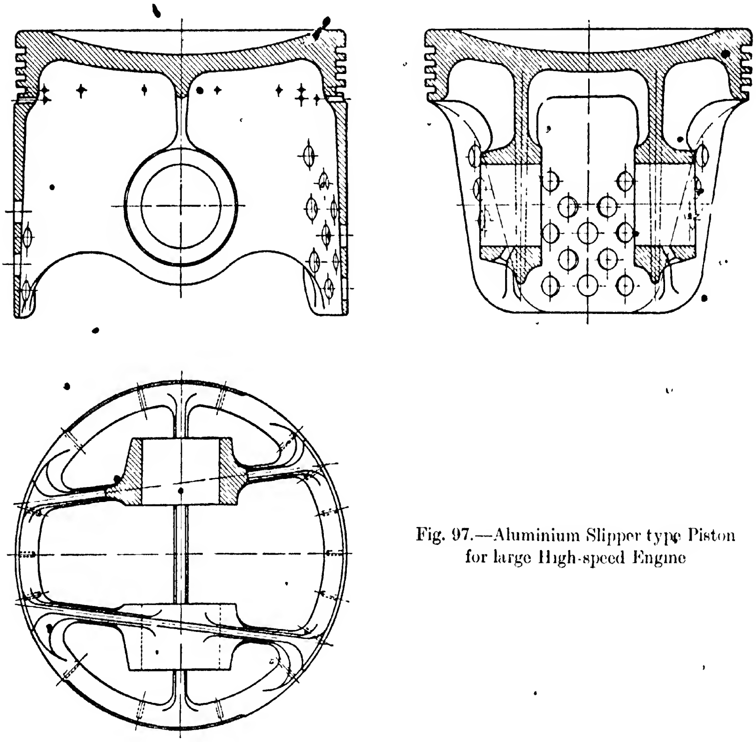


Fig. 97.—Aluminium Slipper type Piston for large High-speed Engine

obtain silent running, for at the end of the compression-stroke the piston is thrust violently from one side of the cylinder to the other. The noise is most apparent when an engine is running slowly on a light load; under such conditions the piston is cool, and therefore the clearance is at a maximum, also the other mechanical noises of the engine are less apparent. Various devices have been experimented with in the endeavour to overcome this troublesome noise. Some designers have used pistons in which the normal clearance is very small and the skirts are slit in order to allow some elasticity

and prevent seizures; others, again, have even gone so far as to introduce springs between the connecting-rod and piston, in order

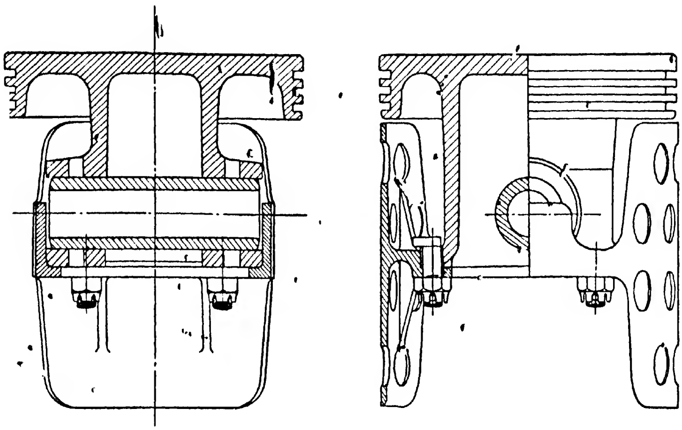
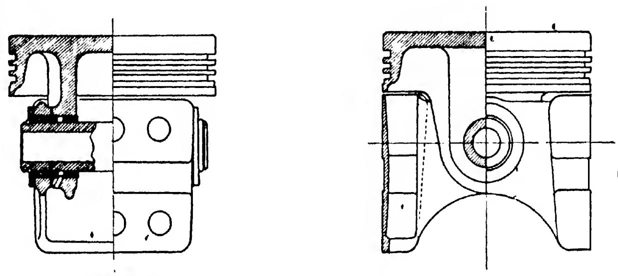


Fig. 98.—Composite Piston with Aluminium Head and Cast-iron Bearing Surfaces rigidly attached

to keep the latter bearing at all times against one wall of the cylinder. Others, again, have adopted the method of fitting the gudgeon pin out of centre in the piston so that the latter tends to tilt about the



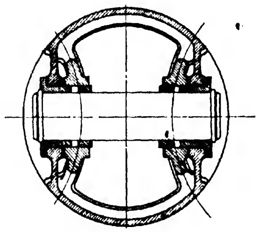


Fig. 99.—Articulated Piston with Aluminium Head and Cast-iron Bearing Surfaces interlocked by Gudgeon Pin

gudgeon-pin centre. This latter method appeared promising at first sight, but on further investigation it was found to be ineffective. The precise effect of offsetting the gudgeon pin in this manner is

illustrated in fig. 100, which shows the position taken up by the piston at various points in the cycle.

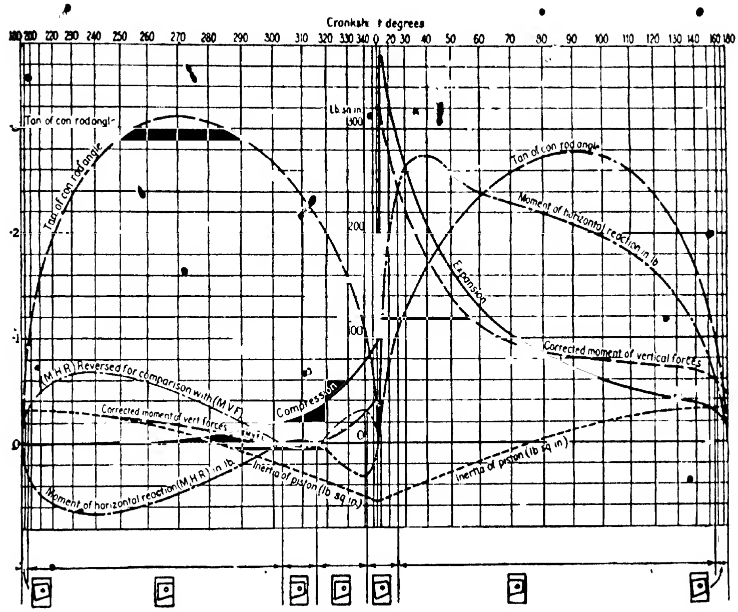


Fig. 100.—Diagram of Forces on Piston with offset Gudgeon Pin, showing effect on clearance

Fig. 101 shows a method patented by the author and applied to a slipper-type piston in which the bearing surface is severed from the crown in order to prevent the direct transmission of heat and so permit of a smaller clearance being used. This arrangement proved very successful in reducing the passage of oil past the piston rings because of the exceptionally free venting of the oil below them, but it did not permit of any appreciable reduction in clearance, for the simple reason that very little heat is transmitted from the crown to the bearing surfaces in any case; in other words, it was found that, at all events in the case of comparatively small engines, the temperature of what may be termed the cross-head portion of the piston was in any case very little in excess of that of the cylinder walls, so that insulating it from the crown had little influence on its temperature, and therefore on its expansion.

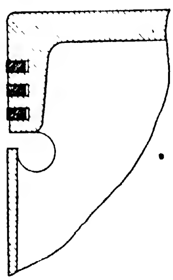


Fig. 101.—Split Slipper Piston

None of the methods described above can be said to have solved the difficulty of piston-knock, or even to have gone very far towards solving it, and the author is inclined to the opinion that where extreme silence is required it is better to employ cast iron for the bearing surfaces in all except very small pistons, in which the clearance may be so small that there is little or no knock in any case, for the noise is dependent upon the absolute, rather than upon the proportionate, clearance. Experience has shown that with all-aluminium pistons fitted to a water-cooled engine of average performance it is necessary to allow a clearance on the bearing surfaces of approximately 0·002 in. per inch of diameter. As a general statement, when the total clearance exceeds from 0·005 in. to 0·006 in. piston knock becomes audible, that is to say, all-aluminium pistons up to 3 in. can be made to run silently, depending upon the lubrication and a number of other minor controlling factors; but above 3 in. diameter it is extremely difficult, if not impossible, to ensure silent running.

Piston Rings.—Generally speaking, piston rings do not call for much comment. With but few exceptions all high-speed internal-combustion engines employ ordinary plain concentric cast-iron rings of the Ramsbottom type. Such rings should always be ground both on the face and sides in order to ensure a close fit in the grooves, and should preferably be hammered, after being split, in order so to stress the material as to ensure a uniform pressure against the cylinder walls. The most important feature to ensure is that there shall be as little clearance as possible in the ring grooves, since this determines largely the amount of oil they will pump. When aluminium pistons are used there is always a tendency for the edges of the "lands" in the piston to be dragged over by particles of grit and so to lock the rings in their grooves. This tendency can, however, be prevented by chamfering slightly the edges of the grooves. This is a small point, but it is one which should not be overlooked, for its neglect has probably done more than anything to prejudice unjustly the use of aluminium for pistons.

Width of Ring.—All reasoning points to the conclusion that piston rings should be made as narrow as possible so long as they are not too fragile to machine or handle. For a given radial thickness, the narrower the ring the less both the friction and the inertia, hence the lower the total pressure against and therefore the wear on the sides of the ring grooves.

Radial Thickness.—The radial thickness determines the

spring tension against the cylinder walls. So long as this is above a certain figure there is no object in increasing the thickness, which merely involves an increase in friction and wear on the cylinder walls.

There is some evidence to indicate that, when working, the rings are pressed out against the cylinder walls by the gas pressure behind them, and that the pressure in the ring groove is something less than the mean pressure of the cycle. This theory is strengthened by the observed fact that when an engine with new rings is motored for several hours without gas pressure in the cylinder the rings do not bed in, but that if run under load they bed very rapidly, particularly the top ring; this may be due to pressure behind the rings, or it may be due to the fact that when run under load the lubrication is less effective. Experience indicates that in a well-made ring a spring pressure of from 5 to 6 lb. per square inch is sufficient, and that any further pressure results merely in extra friction without any compensating advantage. The spring tension required, however, depends to some extent upon the amount of clearance between the "lands" of the piston and the cylinder walls. It is customary and proper to make the clearance of the "lands" such that they will not, under any circumstances, touch the walls of the cylinders. If, however, the clearance is too great, a considerable area of the side of the ring may be exposed to the full fluid pressure, and, as a result, the ring may be pressed so hard against the lower face of the groove that its spring tension will not overcome the friction against the side of the groove; under these conditions the ring will become locked and unable to expand against the cylinder walls.

There is also very strong evidence to show that the top land should always be as deep as possible, in order to provide adequate protection for the top piston ring. In the designs of piston illustrated previously, the top land is in most instances rather too shallow, due to the fact that the available height from the gudgeon-pin centre to the piston crown was, in each case, so restricted as to render the provision of an adequate protecting land impossible.

Cross-head Piston.—All the foregoing remarks refer to the open or trunk type of piston, and more particularly to that form of open piston which has come to be known as the slipper type, a form with which the author has had the most experience.

In the larger sizes of high-speed engine more particularly, the

author prefers to use wherever possible a somewhat different type which is now generally known as the cross-head piston.

In this design the two functions of an open piston—namely, to act as a piston proper and as a cross-head guide—have been separated to a far greater extent than in the case of the slipper type, until it resembles much more nearly the usual steam-engine form.

The piston itself consists of an ordinary flat or concave crown carrying the piston rings and a plain light tubular stem extending from the crown of the piston to below the gudgeon pin. The lower

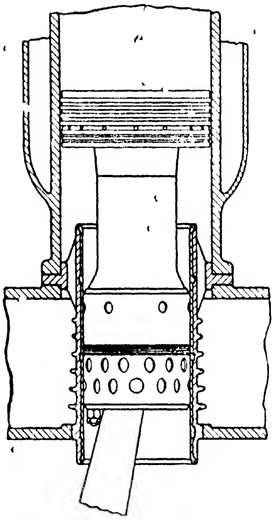


Fig. 102.—Part Section showing Cross-head Piston and Guide

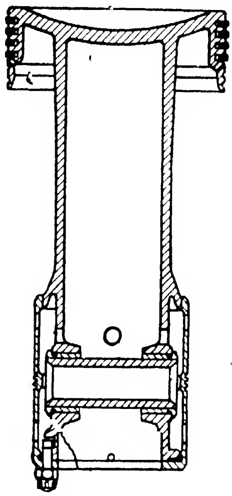


Fig 103.—Section of Piston

portion of this trunk is surrounded by a steel or cast-iron sleeve, which embraces and locates the floating gudgeon pin, and constitutes the only wearing surface. This cross-head sleeve runs in a cylindrical guide, which is spigoted both into the cylinder and the crank chamber, or in some cases into the cylinder only, which is prolonged to accommodate it. A general arrangement of the piston, cylinder, and cross-head guide is shown in fig. 102, from which it will be observed that the crown of the piston serves only to carry the rings and transmit the pressure down the hollow cylindrical stem to the gudgeon pin. It does not bear upon the cylinder walls at all, and therefore requires only just sufficient lubricant to maintain the rings in good condition.

; will also be observed that in this construction the cylinder walls re cut off from all splash lubrication.

In spite of the fact that this piston is ome 30 per cent heavier than the slipper ype, the total friction is little more than 80 er cent that of the slipper piston and only bout 60 per cent that of a piston of the rdinary trunk type.

The construction of the piston is shown in etail in fig. 103, and the photographs reproduced in figs. 104 and 105. The cross-head leeve is an easy push-fit over the lower portion f the stem, and is held in place by four small olts; it is supported by three bearing lands, ne at the middle on the gudgeon-pin centre ne and one at each end.


Fig. 104.—Cross-head Piston

In ordinary commercial engines this sleeve s made of cast iron, but in very high-speed ngines a light high carbon steel sleeve is used. When employing

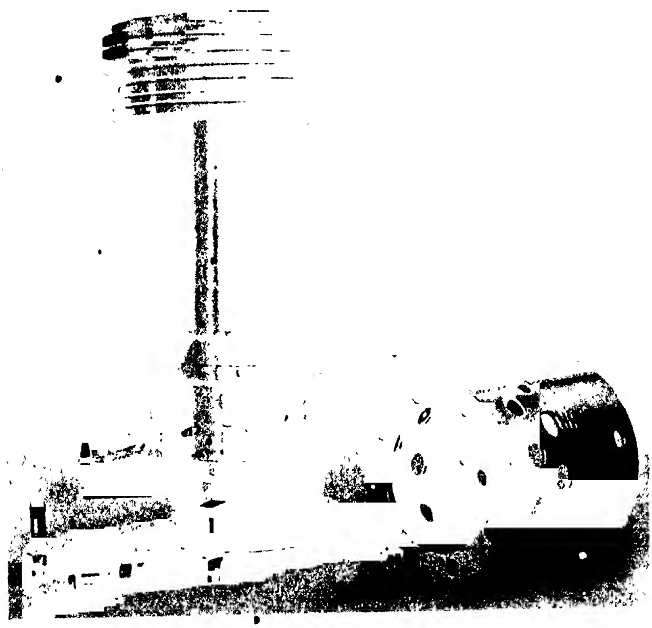


Fig. 105.—Cross-head Piston dismantled

this piston on engines having separate cylinders the guide is spigoted both into the cylinder and crankcase, and the latter is provided with a false top as shown in fig. 102 above. The space between the false top and the base of the cylinder is utilized for the circulation of air round the cross-head guide, which is ribbed for cooling. The air enters at one side and passes out to the carburettor on the other side. A portion of the air is drawn directly round the cross-head guide, and the remaining portion passes between the guide and the cylinder through slots provided for this purpose. On the upward stroke of the piston the air is drawn through these slots at a high velocity, and impinges against the crown and stem of the piston, thus effectually cooling them. On the downward stroke this heated air is discharged again into the chamber surrounding the guides, and thence into the carburettor. By this means the piston and cross-head guide are kept cool and the carburettor air is warmed.

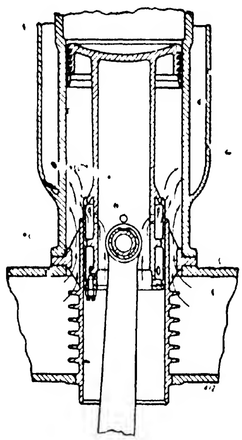


Fig. 106.—Cylinder Lubrication with Cross-head Piston

It is found in practice that the heat abstracted from the piston and cross-head guide is just sufficient for good distribution with petrol of high boiling-point and low volatility. Tests carried out on several of the engines built for tanks with thermometers fitted in the induction piping above and below the carburettor have shown that when running on full load with an atmospheric temperature of 60° F. the air, after passing round the cross-head guides and pistons, entered the carburettor at a temperature of 130° F. On a light load, with consequent reduced air circulation, the temperature of the air entering the carburettor rose to 150° F., and the temperature near the top of the induction pipe to 100° F., which is sufficiently high to check condensation at reduced loads. The free circulation of air through the upper portion of the crankcase tends to keep the lower portion cool, so that no oil cooling is required.

The system of cylinder lubrication is shown in fig. 106. The lower portion of the stem of the piston is provided with a few small holes, and the cross-head sleeve which surrounds it is also provided with a ring of small holes so placed that these holes are uncovered above the guide at the top of each stroke. On the upward stroke of the piston, air is drawn through slots provided in the flange of the cross-

head guide between the guide and the cylinder, and passes at a high velocity around the cross-head sleeve; in doing so, it draws a small proportion of air and oil mist from the holes in the cross-head sleeve, which are in communication with the crank chamber through corresponding holes drilled in the piston stem. The oil issuing from these holes in the form of a mist is picked up by the rush of air and sprayed over the cylinder walls while the piston is near the top of its stroke; the total quantity of oil drawn out in this manner is minute, but it is sufficient for the maintenance of the piston rings. The whole operation is similar to that of a spray carburettor in which the slots in the cross-head guide correspond to the choke tube, and the holes in the sleeve to the jets. The control of the quantity of oil delivered in this manner is governed by the area of the slots and the size or number of holes provided in the sleeve.

It will be seen that, by this means, the lubrication of the cylinder walls is continuous, that oil is only supplied to the cylinder walls in the quantity required by the piston rings, and that oil which has clung to the walls and become partially carbonized does not find its way back into the crankcase. The provision in this manner of an entirely separate system of lubrication to the cylinder allows of the use of unstinted lubrication to all the other working parts without the risk of carbonization of the piston or any tendency to smoke; also, the oil consumption is exceedingly low.

When working with kerosene or high boiling petrols, this type of piston is particularly suitable, for one of the chief troubles with such fuels is that they tend to precipitate upon the relatively cool walls of the cylinder barrel and so to pass down into the crankcase, thus contaminating the lubricating oil, and cause trouble with the bearings. With the cross-head type piston, however, any fuel which may succeed in passing the piston is trapped in the chamber surrounding the cross-head guides, from which it may be drained off before it can do any harm. The quantity of kerosene which, in practice, is drained away from this chamber is often surprisingly great, particularly when working on variable loads, often amounting to as much as from 4 to 8 per cent of the total fuel consumption of the engine, or from three to six times the oil consumption.

The advantages of this type of piston may be summarised as follows :—

(1) The lubrication is under complete control, and is independent of the crankcase lubrication; consequently the oil consumption, the tendency to carbonize both the piston and combustion chamber,

and the risk of oiling up the sparking plugs are all reduced to the minimum.

(2) The piston friction is reduced to little more than half that which obtains with an ordinary trunk piston.

(3) Owing to the fact that the cross-head and guide are relatively cool, and that both are maintained at approximately the same temperature, a very fine running clearance can safely be used, thus ensuring silent running.

(4) Since the piston itself does not bear upon the cylinder walls, an ample working clearance can be allowed without any risk of noise.

(5) The wear on the cylinder walls is reduced to a minimum, since only the piston rings bear against them and there is no side thrust.

(6) The gudgeon-pin being short, stiff and free to rotate, and also being placed in such a position that it receives very little heat from the piston, does not wear perceptibly.

(7) The bulk of the heat from the crown of the piston and from the cross-head guide is utilized to warm the air for the carburettor, and is not transferred to the crankcase.

(8) All the working parts can be lubricated without stint and without any risk of excess of oil reaching the cylinder walls; also, the oil remains clean.

(9) In the event of any fuel condensing on the walls of the cylinder, its subsequent passage into the crankcase can be prevented absolutely.

(10) The restricted lubrication to the cylinder walls greatly reduces any tendency of the piston rings to become carbonized or gummed up.

(11) There is no tendency for the engine to become "gummed up" when cold.

The principal objections to the use of this type of piston are:—

(1) That it increases the height of an engine as compared with the use of an open-type piston, by an amount equal to about two-thirds of the piston's stroke.

(2) That it necessitates an engine designed specifically for its use, and unless separate cylinders are used it introduces difficulties in the way of alignment between the cross-head guides and the cylinder bore. These difficulties are not, however, insuperable, as is illustrated by the engine shown in fig. 107, which shows the application of cross-head pistons in the case of the engines manufactured by Messrs. Peter Brotherhood, Ltd., for tractors and marine work, in

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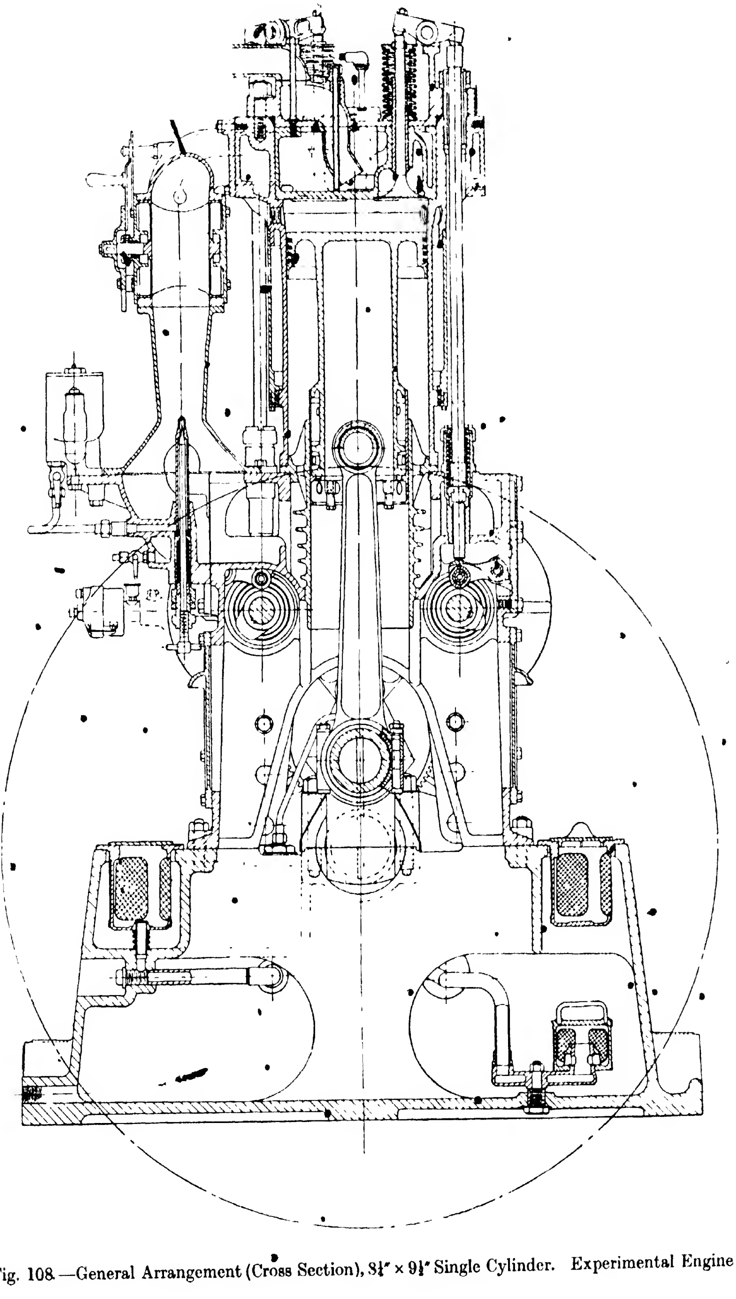

Fig. 108.—General Arrangement (Cross Section), 8¼″ × 9½″ Single Cylinder. Experimental Engine

which the four cylinders and the upper half of the crankcase are cast in one piece.

(3) That it increases slightly both the cost and weight of an engine.

These objections are, in the author's opinion, easily outweighed by the advantages to be gained, more especially in the case of large engines, such as that shown in fig. 108, or in the case of engines using kerosene or petrol with a high final boiling point. In any case their use makes for a higher mechanical efficiency, silent running, and reduced cylinder wear and oil consumption.

CHAPTER X

ENGINES FOR ROAD VEHICLES

Before dealing with specific examples of motor-vehicle engines it will be well to review briefly the duties which these engines have to perform and to note along what lines further development is likely to extend.

Some twenty-five years ago the designers and manufacturers of motor-vehicle engines had need to concentrate the whole of their attention upon the one crucial question of producing engines which would run for a reasonable period, under wholly novel conditions, without serious breakdown. This great problem overshadowed all others, and the rapidity with which it was tackled and overcome is one of the great triumphs of modern mechanical engineering. Within a space of less than ten years the motor-vehicle engine emerged from the stage of a fickle and wayward, but very fascinating, toy into a thoroughly reliable machine. Once its reliability had been established, and its charm largely exchanged for utility, the subsequent developments were mainly in the direction of refinement and increased power.

In order to provide a more uniform turning movement, and to reduce vibration, and therefore noise, the number of cylinders was increased from one to four and even to six.

The next developments were in the direction of securing greater silence in operation; these took the form of improvements in the valve gear and the elimination as far as possible not only of vibration of the engine, as a whole, but also of the vibration of the individual members of the engine.

At the same time the available speed range has steadily increased. Since owing to the inherent irregularity of torque in any type of four-cycle engine the speed range cannot usefully be reduced below a certain minimum, progress has taken the form of extending the upper end of the speed range. Developments in this direction have been much stimulated by the method of basing the taxation, of

pleasure vehicles at least, upon the diameter of cylinder used. This basis of taxation has served well, but would now be more useful if it were based on the total cylinder capacity.

With the extended use of motor-vehicles for purely utility purposes, and with the increasing cost of fuel, the most needed developments at the present time are in the direction of fuel economy, a direction in which there is still ample scope for improvement.

The motor-vehicle engine of to-day is called upon :—

(1) To be silent under all conditions of operation.

(2) To be self-contained and as automatic as possible.

(3) To have as wide a range of speed as possible.

(4) To accelerate rapidly from any speed ; in other words, it must instantly develop its maximum torque when called upon to do so, irrespective of engine speed.

(5) To maintain a high torque at low speeds, and to do so without detonation or "pinking."

(6) To be reasonably economical in fuel at all loads, and more particularly at its average load factor of from 25 per cent to 40 per cent maximum torque.

The performance of any motor-vehicle engine must be considered in reference to the vehicle to which it is fitted. We will therefore examine briefly one specific instance, namely :—

A light pleasure car weighing, fully loaded, including passengers and equipment, 3500 lb., and fitted with a wind-screen and hood. We will assume that the transmission gear is of an efficient type, that the unsprung weight is low, the weight well distributed, and that the chassis generally is as well designed throughout as the present state of the art will permit.

Unfortunately, very little accurate data is available as to the exact power required to propel a motor-vehicle at different speeds over average roads. Professor Riedler in Germany, and Chase and James in America, have made and analysed a number of dynamometer tests with the rear wheels of the car resting on rollers, but these do not always reproduce the conditions exactly. For information on this point we are compelled to fall back to a large extent upon tests carried out with accelerometers and upon accumulated experience based upon the known performance of the same engine, both on the test-bed and on the road—the latter method, though very unscientific and largely empirical, is probably the most accurate at the present time. The curve in fig. 109 gives to the nearest approximation the brake-horse-power required at the engine

flywheel to propel a 3500 lb. 4-seater car at speeds up to 80 miles per hour. It includes rolling-resistance, windage, transmission losses (on direct drive), and all other incidental losses such as wheel slip, hysteresis losses in the tyres, &c. Though purely empirical it is probably reasonably correct. For such a car the minimum size of engine of normal side-valve type which will give reasonable acceleration and hill-climbing capacity will be one of two-litre cylinder capacity, while for real comfort a three-litre engine will be preferable.

We will consider both cases and assume that the engines are of

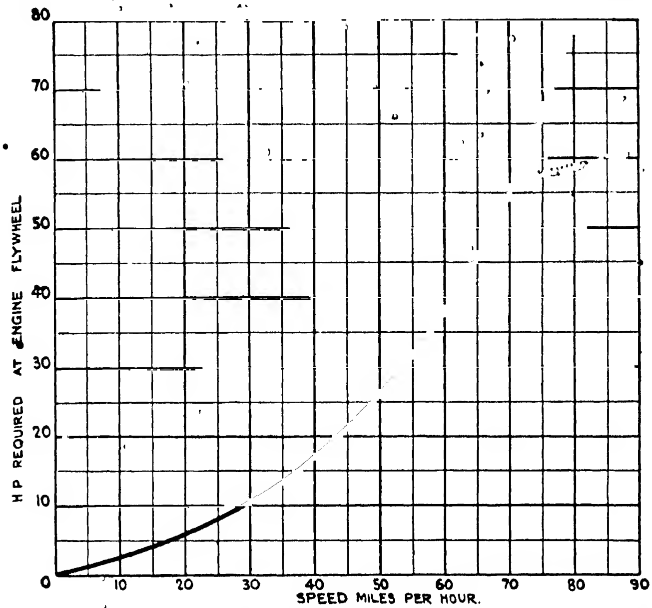

Fig. 109.—Power required at Engine Flywheel to propel Car of 3500 lb. gross Weight on the Level at varying Speeds over average Road Surfaces

the normal side-by-side valve type fitted with as efficient a form of combustion chamber as this type will permit of. Further, we will assume that the engines are designed with a view to low cost of production and ease of upkeep, that they have a reasonably low compression ratio, viz. 4·6 : 1, to render them capable of using inferior fuel without detonation, and generally that they are of a thoroughly orthodox type, but as efficient as possible without resorting to the use of overhead valves or to any features involving either increased cost of production, or labour in upkeep.

Figs. 110, 111, and 112 show the brake horse-power and general

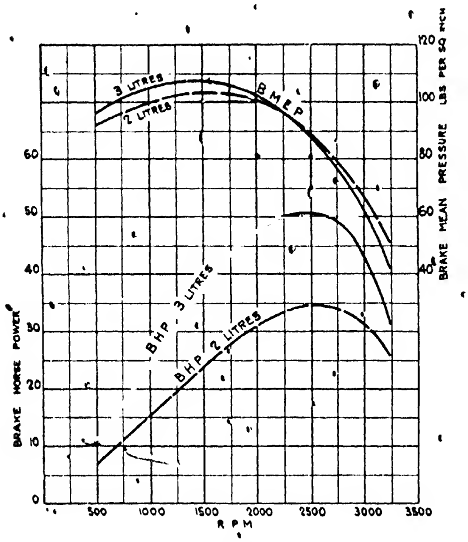

Fig. 110.—Brake Horse power and Brake Mean Pressure of normal Two- and Three-litre Engines

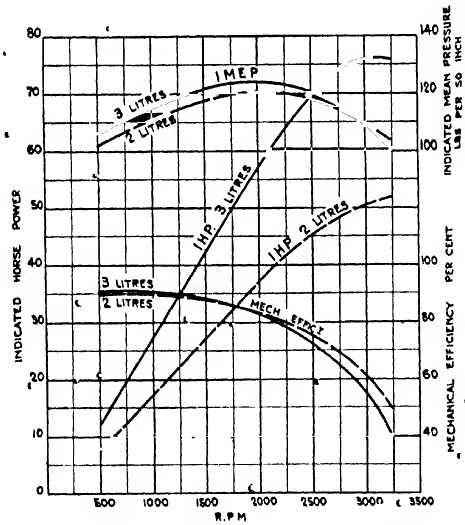

Fig. 111.—Indicated Horse power, indicated Mean Pressure, and Mechanical Efficiency

performance curves which should be obtainable from such engines of two- and three-litre cylinder capacity.

Next we will assume that, in both cases, the top gear ratio is such that the maximum speed on the level is obtained when the engine is running at a speed slightly in excess of that at which maximum power is developed—this is always desirable, both on the grounds of acceleration, and in order to reduce the hysteresis losses due to irregularity in the turning moment. We will assume that three speeds are used and that the ratios in the gear box are such that the second speed is 70 per cent and the third 33 per cent. of the top or direct drive. From a comparison of the power curves of the two engines and the power required on the level, as shown in fig. 109, we find that the most suitable gear ratios for the direct drive are those which give a car speed of 20 miles per hour at 1100 R.P.M. in the case of the two-litre, and at 880 R.P.M. in the case of the three-litre engine. If now we plot the power

curve of the two engines against fig. 109 in the above ratio, as shown in fig. 113, we find that the two-litre engine will give a maximum speed on the level of 54·5 miles per hour and the three-litre of 66 miles per hour. The margin of power at any speed over and above that required to propel the car on the level may be termed the excess power available for hill-climbing or acceleration.

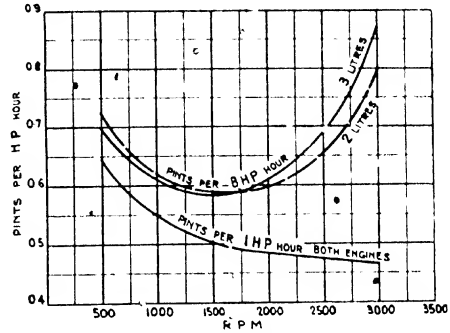

Fig. 112.—Fuel Consumption at full Torque: Pints per Indicated and per Brake Horse-power Hour

Figs. 114 and 115 show the excess-power curves for the two engines on the three gears, assuming an efficiency as compared with top gear of 95 and 97 per cent respectively for the first and second speeds. Strictly speaking, the relative gear losses will be less in the case of the larger engine, but the difference is small and hardly worth taking into consideration. Figs. 114 and 115 show also the gradient in terms of per cent which the car will climb on each

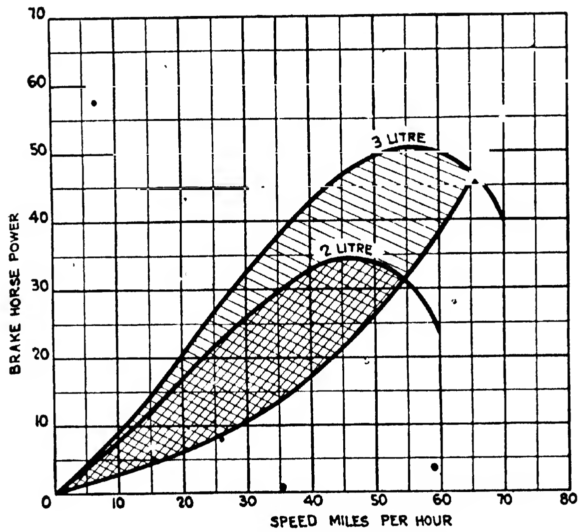

Fig. 113.—Excess-power Curves for Two- and Three litre Engines on Top Gear

gear and the speed at which it will climb it without gain or loss in speed.

From the curves shown in these two figures it will be seen that

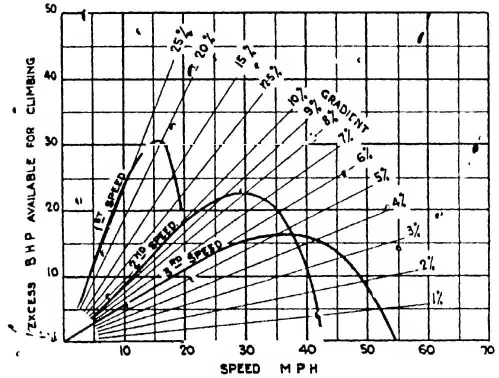

Fig. 114.—Hill-climbing Capacity, Two-litre Engine

the maximum gradient which the three-litre engine will climb without gain or loss of speed on its third or top gear is one of 8 per cent, on its second speed the maximum gradient is about 12·5 per cent, and on its bottom speed about 31 per cent. It will be noted

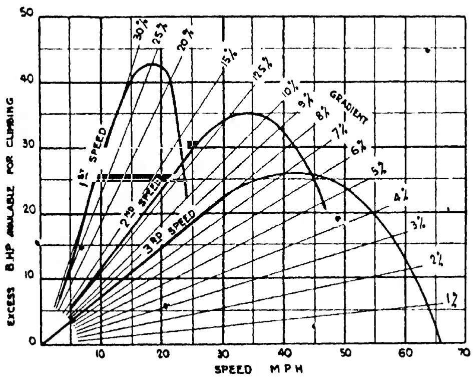

Fig. 115.—Hill-climbing Capacity, Three-litre Engine

that on a gradient of 6·2 per cent the maximum speed will be the same on either top or second speed, namely, 45 miles per hour.

In the case of the two-litre engine, the maximum gradient which the car will climb at a uniform speed on top gear is one of 6 per cent, on second speed 10 per cent, and on bottom speed about 25·5 per cent. For maximum speed in this case, gear should be changed

from top to second when the gradient exceeds 4·7 per cent, or when the speed has dropped to 38 miles per hour.

Fig. 116 gives the rate of acceleration of the car with two-litre engine, from any speed, and on the three gears, assuming that the carburation and distribution are such that the engine will respond instantly and exert its maximum torque immediately the throttle is opened—a condition, however, which is seldom reached in practice. The foregoing curves show the general performance of the car as regards ultimate speed, acceleration, and hill-climbing capacity.

We have next to consider the question of fuel consumption and the factors which control it. For this purpose we will assume that the car will be running always on its top gear, and we will examine

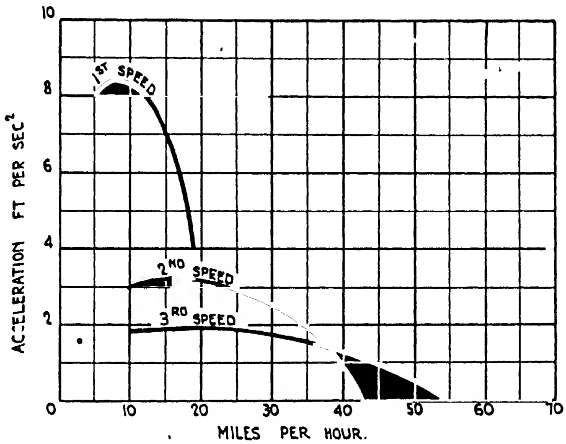

Fig. 116.—Acceleration on various Gears, Two-litre Engine

the speed range between 10 and 40 miles per hour, which covers the range of *average* speed such a car will maintain. For simplicity we will assume also that the car is running on a level road, though, in so far as fuel consumption is concerned, it makes comparatively little difference whether the road is level or undulating provided the gradients are well within the limits which the car can negotiate on top gear and that the average speed is not too low. Although, when coasting, one does not recover what is lost in climbing, yet this is very nearly compensated for by the more favourable load factor when pulling uphill.

Figs. 117 and 118 show the load factor in the case of the three- and two-litre engines at speeds ranging from 10 to 40 miles per hour, and the fuel consumption in terms of pints per B.H.P. hour at the

corresponding load factors and speeds. These figures are deduced from the mean of a large number of test results upon several engines

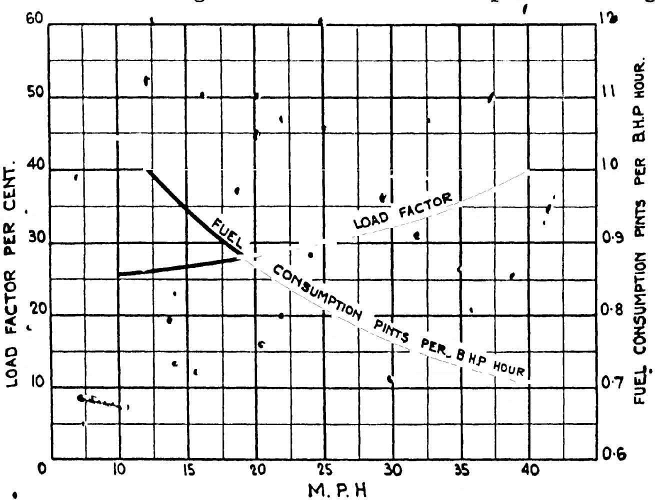

Fig. 117.—Load Factor and Fuel Consumption in Pints per B.H.P. Hour when running on 1 Level at average Speeds of from 10 to 40 Miles per Hour. Three-litre Engine

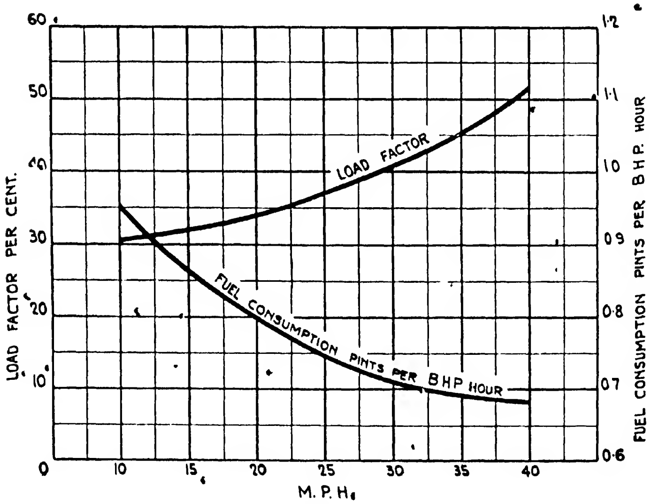

Fig. 118.—Load Factor and Fuel Consumption in Pints per B.H.P. Hour when running on the level at average Speeds of from 10 to 40 Miles per Hour. Two-litre Engine

of the size and class under consideration with carefully adjusted carburettor and ignition settings and a reasonably good distribution system. In fig. 119 is shown the fuel consumption in terms of miles

per gallon for the two engines at average car speeds ranging from 10 to 40 miles per hour.

With normal carburation the consumption per mile is about 7 per cent greater with the larger engine at an average speed of 20 M.P.H., but the discrepancy becomes less as the average speed increases. With perfect carburation and distribution, &c., the discrepancy will become less, and at the higher mean speeds the larger engine will show, with the gear ratios selected, an actually greater fuel economy than the smaller one. In either case the larger engine will, in fact, make a better showing if the road is hilly or undulating, for it will then be able to negotiate gradients on top speed which, in the case of the smaller engine, might necessitate a change of gear.

There is another factor which also exerts a still more powerful

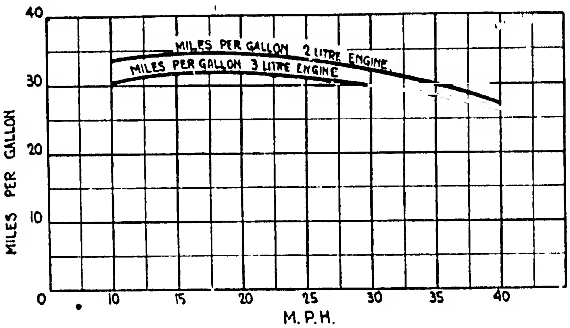

Fig. 119.—Fuel Consumption in Miles per Gallon at average Speeds varying from 10 to 40 Miles per Hour

influence upon fuel economy than carburation and distribution, and indeed upon the whole performance of the car, namely, the mechanical efficiency of the engine. This becomes the more important because of the very low load factor at which the engine operates. In the example shown, a fairly high mechanical efficiency has been assumed such as would be obtainable with light reciprocating parts and careful mechanical design. The average motor-car engine with cast-iron pistons and often excessive and ill-disposed bearing surfaces will not show by any means so high a mechanical efficiency.

It is perhaps worth while to consider the case when the car is travelling on the level at a mean speed of, say, 25 miles per hour and observe the influence of the mechanical efficiency of the engine upon fuel consumption. This speed calls for an expenditure of 8 B.H.P. at the flywheel of the engine and corresponds to an engine speed of 1100 R.P.M. in the case of the three-litre and 1375 R.P.M.

in the case of the two-litre engine: at these speeds the mechanical efficiency of the two engines has been taken as 90·5 per cent and 88·5 per cent respectively on full torque, but since the internal friction of the engine is independent of the torque, at 8 B.H.P. the mechanical efficiency will have fallen to 73·5 per cent and 73 per cent respectively. If now the mechanical losses were doubled owing to poor mechanical design, all other conditions remaining the same, the mechanical efficiency when driving the car on the level at an average of 8 B.H.P. will become only 58·2 and 58·6 per cent respectively, and the fuel consumption per B.H.P. hour will have increased from 0·81 and 0·745 to 1·02 and 0·93 pints per H.P. hour at the same load factor, but in fairness we must allow for the fact that, owing to the poorer performance as a whole, the load factor will be somewhat higher. If we take this into account we find that the consumption at a mean speed of 25 M.P.H. will be approximately 0·97 and 0·89 pint per B.H.P. hour, corresponding to a fuel consumption in terms of miles per gallon of 25·8 and 28·1 as against 31 and 33·6 for the three- and two-litre engines respectively. From these figures it will be seen that the gain in fuel economy to be obtained by a limited and perfectly possible improvement in mechanical efficiency is a very substantial one. Further, a gain in mechanical efficiency will influence not only the fuel economy but also the speed and hill-climbing capacity of the car throughout its whole working range. From such considerations we are justified in assuming that of the available scope for improvement the most important is that of reducing as far as possible the internal friction losses of the engine, and next in importance are improvements in carburation and distribution.

Unlike engines for other purposes, we may regard the pleasure-car engine as one which will never be called upon to develop high power, except for very short periods, and we have shown that the average load factor under normal running conditions is about 30 to 40 per cent in the case of the engines under consideration. Expressed in other terms, the average power required at the engine flywheel to propel a touring motor-car under normal conditions at an average speed of 25-30 miles per hour is approximately 7 H.P. per ton (unladen), while with even the most reckless driving it is almost impossible on any English main road to average 15 H.P. per ton, altogether irrespective of the maximum power of the engine.

In this connection it is interesting to note that from careful observations of fuel consumption made during the practice runs for the Isle of Man Tourist Trophy Race in 1922, the average horse-

power developed by the Vauxhall racing cars during their fastest laps, when they averaged considerably over 60 miles per hour round a perfectly clear and very hilly course, was certainly less than 50 B.H.P., even assuming that they were using the most economical carburettor setting. In view of the fact that these engines were capable of developing well over 120 B.H.P., and that the cars were naturally driven at the highest possible speed consistent with barely reasonable safety, it appears rather surprising that so small a proportion of the available power could be utilized. It shows that even when roads are cleared of all traffic and when the driver is relieved of all responsibility so far as other road-users are concerned, when he is both highly skilled and prepared to incur considerable personal risk, he is still restricted, by road conditions, to utilizing more than about 40 H.P. per ton.

Most cars at the present day show an unduly high fuel consumption, and this is to be accounted for by

(1) The mechanical efficiency of the engine being usually very low; in the one application above all others where it should be as high as possible.

(2) The form of the combustion chamber being generally inefficient, due to lack of turbulence.

(3) Defective carburation and distribution, more particularly the latter.

Recent development has been confined almost solely to the addition of various refinements, to the elimination of noise and general smoothness of operation; such lines of development are, of course, very proper, but there is a tendency for the economic fact, that the efficiency of a vehicle as a whole lies in the number of ton-miles it will run per gallon, to be overlooked. In too many cases fuel efficiency appears to have been forgotten entirely in the search for silence, in the better-class cars, and for low cost of production in the cheaper varieties. The author uses the word forgotten advisedly as against forgone, for, as it has been shown in previous chapters, fuel economy is largely a question of design and can usually be attained without adding to the cost and without the loss of other desirable features. The history of engine development has been much the same in all classes of mechanical engineering—first a struggle to attain mechanical reliability, during which stage the engine is a fascinating toy; this is generally followed by a period of intense rivalry in detail refinement to the neglect of other considerations; finally the inexorable laws of economy insist that

attention shall be concentrated on what is really the final test, namely, the amount of work an engine will do on a given quantity of fuel and on a given weight and cost of material. In the case of the motor vehicle we are probably passing from the second to the third stage of development and are beginning to realize the absurdity of, for example, loading the engine at all times with a heavy burden of frictional losses often merely for the sake of getting it to run a little slower and a little quieter when idling. As in the case of all new developments which fall into the hands of a lay public, fashion plays

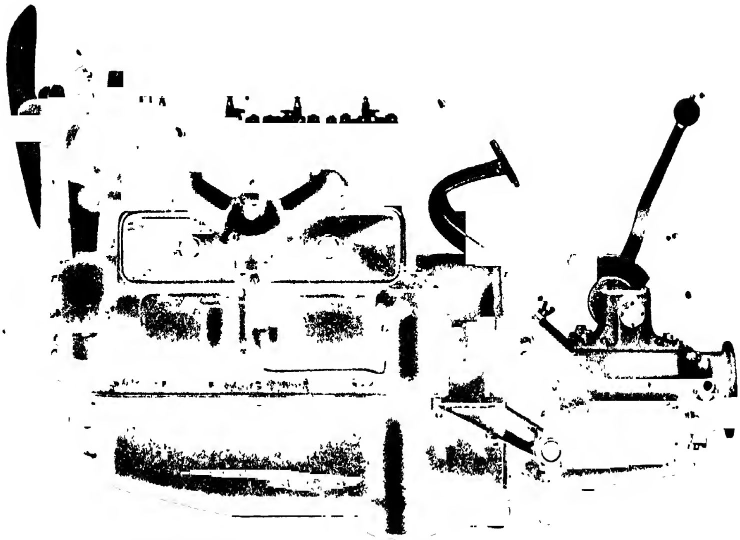

Fig. 120.—14-H.P. Vauxhall Engine

a predominant part, and fashion to-day calls for refinement in detail to the neglect of all other considerations. Ultimately utility will call for economy in operation, and the attention of designers will be concentrated upon reducing mechanical losses and improving distribution.

The 14-H.P. Vauxhall Engine.—The 14-H.P. Vauxhall engine illustrated in figs. 120, 121, 122, and 123 has been designed by Mr. C. E. King, Chief Engineer of the Vauxhall Company, to whose kindness the author is indebted for leave to publish the following particulars:—

It may be regarded as a typical example of a modern pleasure-car engine of the best type, designed to meet as far as possible the

dictates of fashion, and at the same time to show a performance both as regards power and fuel economy considerably above the usual average of engines of its class. It has four cylinders each of 75-mm. bore by 130-mm. stroke, giving a total cylinder capacity of approximately 2·3 litres, and is designed to drive a five-seated open touring car weighing complete with passengers and usual equipment about 3200 lb. It develops a maximum of 43·5 B.H.P. at a speed of 2600-2700 R.P.M.

Particular care has been taken to reduce, as far as possible, the internal friction losses, and also to obtain an efficient form of com-

Fig. 121.—14-H.P. Vauxhall Engine

bustion chamber, with the result that the power output and efficiency are both very considerably greater than that of the average side-valve engine, particularly so at reduced loads. The details of the design are shown in figs. 122 and 123, from which it will be seen that the four cylinders are cast in one block separate from the crankcase and with a common detachable aluminium cylinder head, the combustion chamber of which is as shown in fig. 122.

The crankshaft is carried in three white-metal lined bearings and is drilled for forced lubrication to all main and crankpin bearings. The pistons are of aluminium of the slipper type, but having a complete ring formed at the base of the slippers. The gudgeon pins float freely, both in the connecting-rods and pistons, and are

located endwise by means of circlips and washers. The total reciprocating weight of each line is 1·75 lb., while the rotating weight of the connecting-rod big end is also 1·75 lb. The inlet valves have a port diameter of 1·4 inches with a lift of 0·35 inch, and the exhaust valves a port diameter of 1·31 inches with the same lift. All valves are operated by means of push rods having curved slippers.

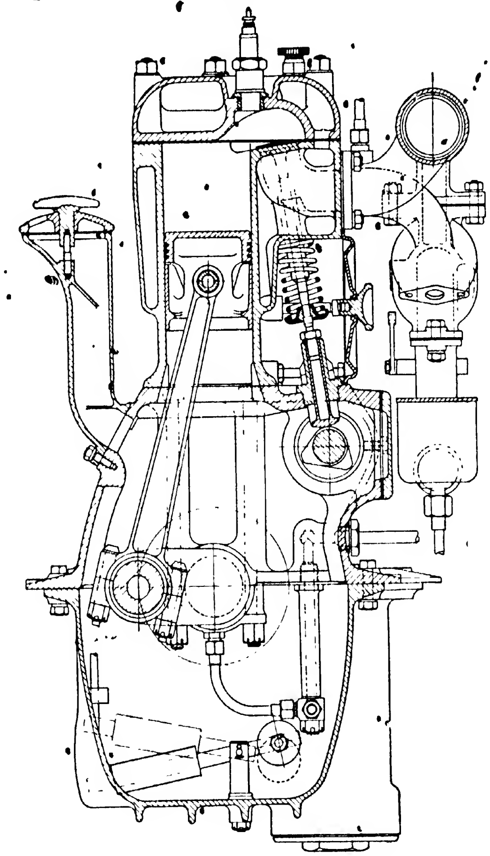

Fig. 122.—Arrangement of Cross-section, 14-H.P. Vauxhall Engine

The crankshaft has a diameter of 1·75 inches throughout, the widths of the several bearings being:

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Forward end	...	...	2·4 inches.
Centre	... ...	...	2·1 ,,
Flywheel	... ...	...	2·65 ,,
Connecting-rod	...	...	1·75 ,,

The ratio, therefore, of the piston area to the projected crankpin area is as 2·22 : 1, so that the load factor on the crankpin bearing is a very light one.

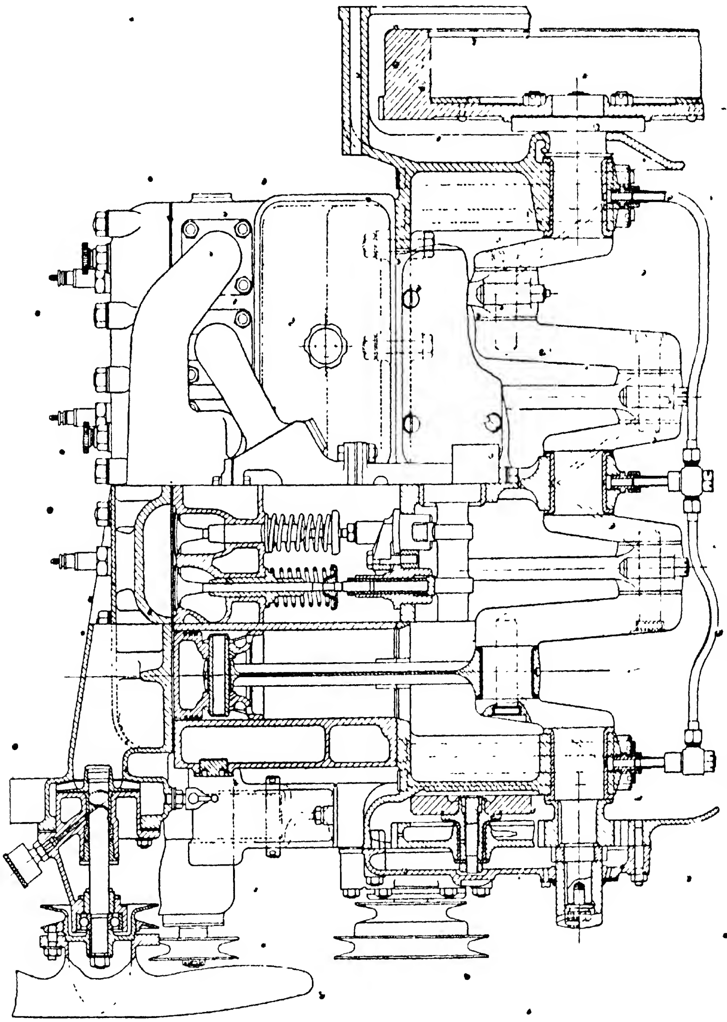

Fig. 123.—Arrangement of Longitudinal Section of 14-H.P. Vauxhall Engine. 75 × 130 mm

The general performance of this engine, with a compression ratio of 5·1 : 1 and with wide-open throttle over a speed range from 750 to 2750 R.P.M., is shown in fig. 124, from which it will be seen that a brake mean pressure of 108 lb. per square inch is obtained at a

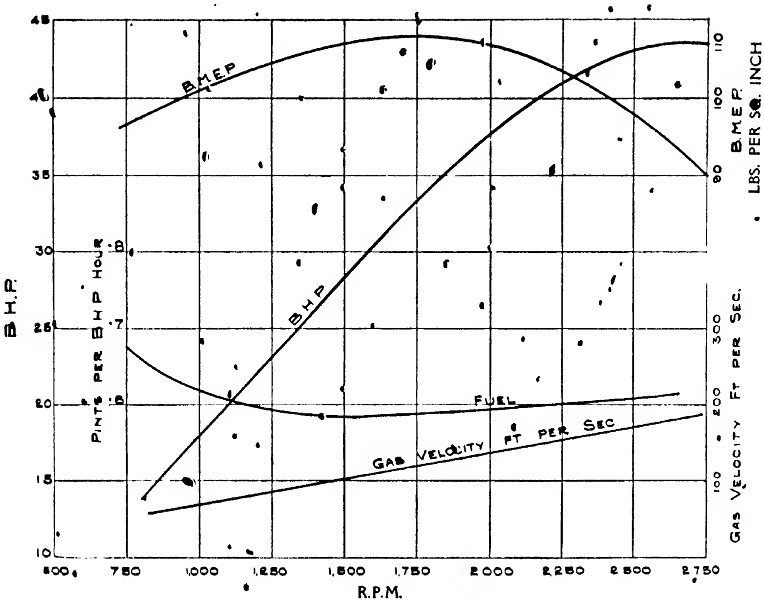

Fig. 124.—14-H.P. Vauxhall Engine. Gas Velocity, Brake Horse-power, Brake Mean Pressure, and Fuel Consumption Curves

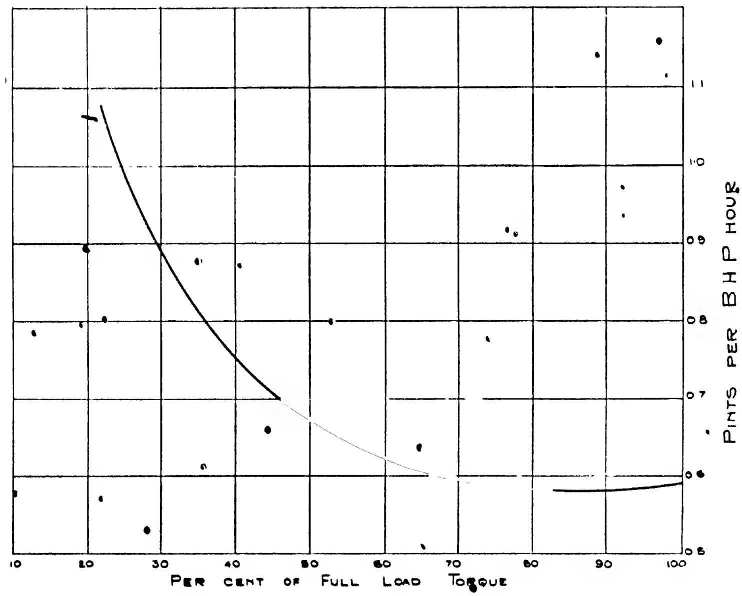

Fig. 125.—14-H.P. Vauxhall Engine. Fuel Consumption various per cent of full Load

speed of 1750 R.P.M. corresponding to a gas velocity through the inlet valves of about 120 ft. per second, the gas velocity through the induction pipe at this speed being about 175 ft. per second; this relatively high velocity is maintained in the induction pipe in order to keep the liquid particles of fuel in suspension even at reduced loads.

Fig. 125 shows the fuel consumption at a speed of 1600 R.P.M. when the power is controlled by throttling, from which it will be seen that even at 50 per cent full-load torque the consumption is less than 0·7 pint per B.H.P. hour.

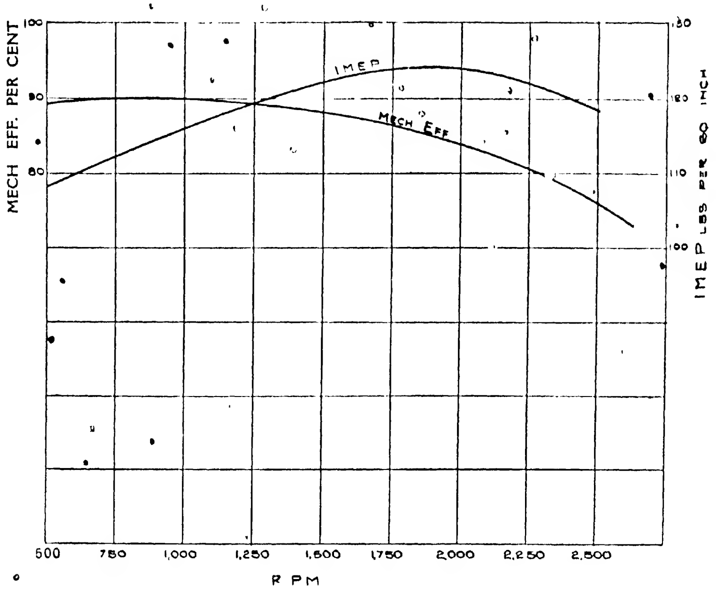

Fig. 126.—14-H.P. Vauxhall Engine. Indicated Mean Pressure and Mechanical Efficiency

No data is available as to the mechanical efficiency of this engine, but it can be estimated fairly accurately from the general design and checked back from the measured fuel consumption at reduced loads. From such deductions it would appear that the mechanical efficiency and indicated M.E.P. are as shown in fig. 126, from which it will be seen that the indicated mean pressure reaches a maximum when the gas velocity through the inlet valves is about 150 ft. per second. The indicated fuel consumption would probably reach its minimum at or about this speed, but, unfortunately, it is clear from the throttle curve that the carburettor was set to give an over-rich mixture at full throttle, so that there is no real evidence available.

The engine drives the car through a three-speed gear-box giving ratios which correspond with road speeds of 6·25, 12·5, and 21 miles per hour at 1000 R.P.M. Its maximum speed on the level is a little short of 60 M.P.H., corresponding to an engine speed, without wheel slip, of 2850 R.P.M., and its consumption averages 30 miles per gallon at an average speed of 25 M.P.H. and 29 miles per gallon at 30 M.P.H.

Sleeve-valve Engines.—For motor-car engines where the need for silence is great and at the same time where, owing to the low average power factor, the heat flow is small, the use of sleeve valves in place of the ordinary poppet valves appears very attractive. Such valves have the following advantages :—

(1) Their action is, or should be, noiseless.

(2) Their employment permits of the use of the best possible form of combustion chamber with the sparking plug centrally situated, hence the indicated efficiency should be high and the tendency to detonate at a minimum.

(3) They require less attention than poppet valves and cannot readily be thrown out of tune by misuse.

The objections are :

(1) That the heat flow to the cooling water is necessarily somewhat restricted, though this is not of much moment in the case of motor-car engines, more especially when a single sleeve is used.

(2) Unless the sleeve be given an abnormally long stroke the effective port area is necessarily restricted.

(3) The sleeve or sleeves, having a large rubbing surface, necessarily entail a higher friction loss, more particularly when a long stroke is used.

(4) It is possible only to operate the sleeve from one side unless the whole of the operating mechanism be duplicated, which involves excessive mechanical complication and introduces grave difficulties in the way of mechanical synchronization.

In a four-cycle engine the sequence of operation is such as cannot be fulfilled by a plain reciprocating sleeve, hence it is necessary either to employ two concentric reciprocating sleeves, as in the Daimler Knight engine, or a single sleeve with a combined reciprocating and rotary motion, as in the Burt engine. A plain rotating sleeve is unsatisfactory, since a reciprocating motion of some sort is essential to prevent scoring of the sleeve and cylinder wall. It is essential also that the whole of the inner surface of the sleeve shall, at every cycle, be scraped either by the piston or the fixed cylinder head,

in order to prevent the formation of shoulders due to wear or carbon deposits, which would prove fatal to their operation.

The use of two concentric and reciprocating sleeves has the advantage that their mechanical operation is somewhat simpler, but it is very difficult to see what further advantage they can possess. The chief fault of the sleeve-valve engine, namely, the difficulty of disposing of the heat from the piston, is greatly accentuated when

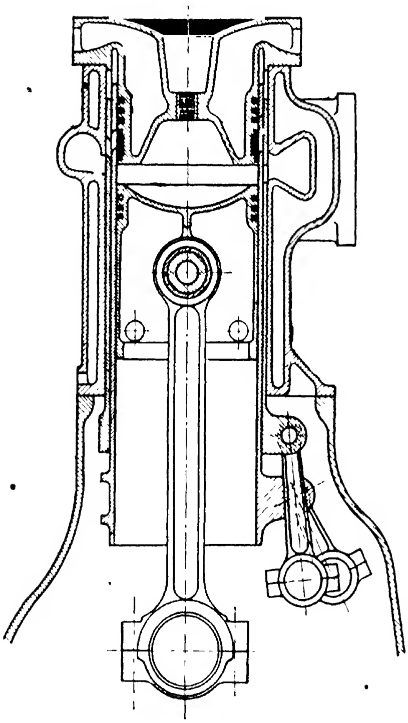

Fig. 127

two sleeves are used, as also the friction loss, which is no small item and a particularly objectionable one where the load factor is light, as in a motor-car engine.

In fig. 127 is shown diagrammatically the operation of a double-sleeve engine, from which it will be seen that the sleeves are actuated from a half-speed crankshaft connected by short rods to points at the side of each sleeve.

Figs. 128-131 show various alternative methods adopted by Burt for operating a single-sleeve valve. The method shown in fig. 128

is that used in the Picard Pictet cars and is, in the author's opinion, attractive from a mechanical point of view, but it is necessarily somewhat costly. In this arrangement two half-speed crankshafts are employed and the sleeve is operated from the centre of

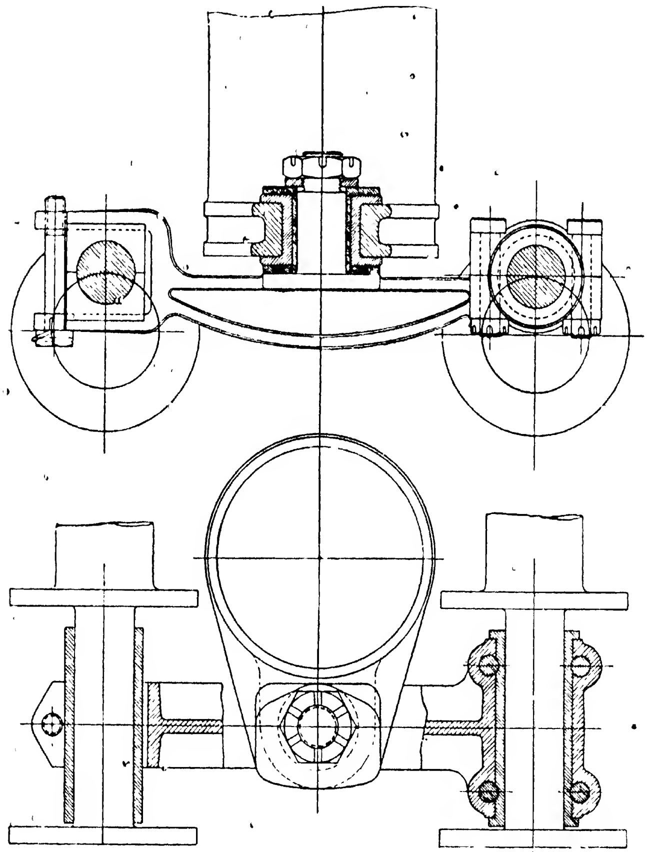

Fig. 128

a coupling-rod connecting these shafts. It will be seen that one end of the coupling-rod is connected directly to one crankpin and the other to a second crankpin, but through the medium of a sliding block with a small amount of end play. The sliding block allows for any slight errors in synchronism as between the two half-speed shafts.

Fig. 129 shows a similar method of operation in which only one half-speed shaft is used. This is said to work well in practice, but is clearly inferior mechanically to that shown in fig. 128.

Another very attractive form is that shown in fig. 130, in which a ball-and-socket joint is used. This form has the advantage of being considerably lighter and more compact;

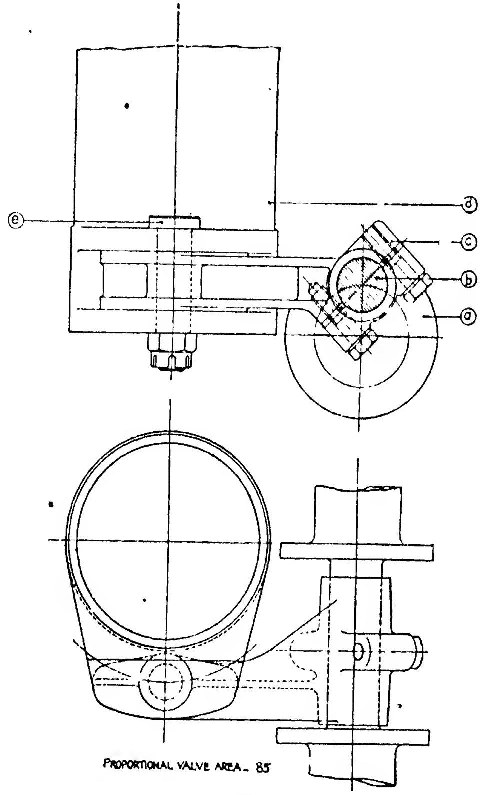

Fig. 129

also it is probably the least expensive and the most accessible. Fig. 131 shows an earlier form used in the Argyll cars, in which a reciprocating plunger is used in place of the ball socket. As in the forms 128, 129, and 131, this recessitates the use of a pin in the sleeve, and therefore both increases the radius of operation and the effective weight of the sleeve.

The author is greatly indebted to Mr. Burt for the following information and particulars as to the determination of port areas, &c., when a single sleeve is employed.

Calculation of Ports.—The special shape of port is adopted to give a maximum area of opening with the minimum of sleeve travel. Fig. 132 shows typical ports a being the ideal shape, b the same port

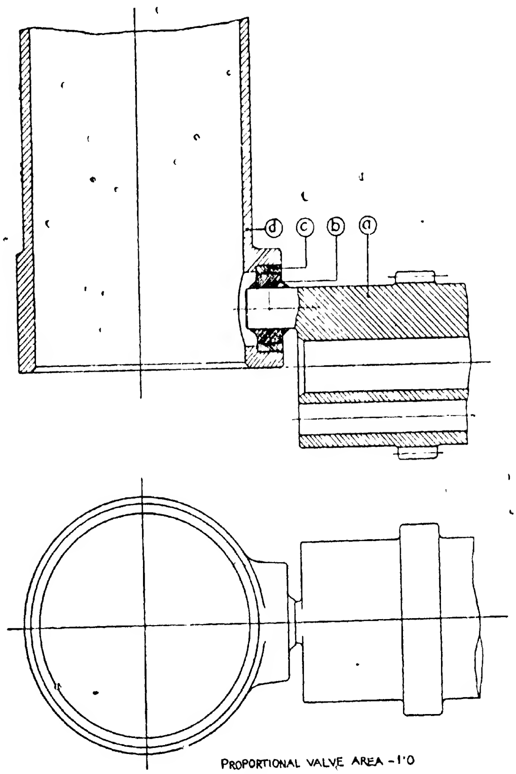

Fig. 130

with corners rounded off to avoid interference when sleeve-valve port is passing down between two cylinder ports. The straight flank port c is usually adopted, as it is a better manufacturing proposition, although slightly smaller in area than type b, for a given valve-shaft stroke.

It is necessary to fix the following particulars before calculating single-sleeve valve ports:—

A = Arrangement and number of ports.
D = Outside diameter of sleeve valve in inches.
C = Distance from axis of sleeve valve to axis of pivot-pin or
ball-and-socket coupling in inches.
T = Throw of sleeve crank in inches.
V = Engine timing.

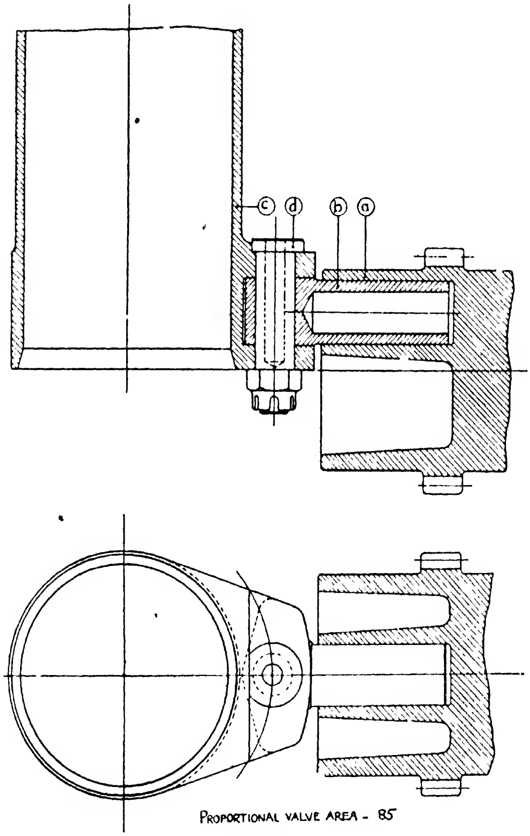

Fig. 131

The greater the number of ports the smaller the sleeve-valve crankthrow for a given area of valve opening, thereby reducing the over-all dimensions of valve-driving mechanism and height of engine, but complicating the coring of water-passages in the cylinder castings and increasing the port cutting time. The fewer the number of ports for a similar area the greater the valve crankthrow and

over-all dimensions, but the coring is simplified with an attendant reduction of port cutting time.

The maximum inlet opening areas obtainable with various port

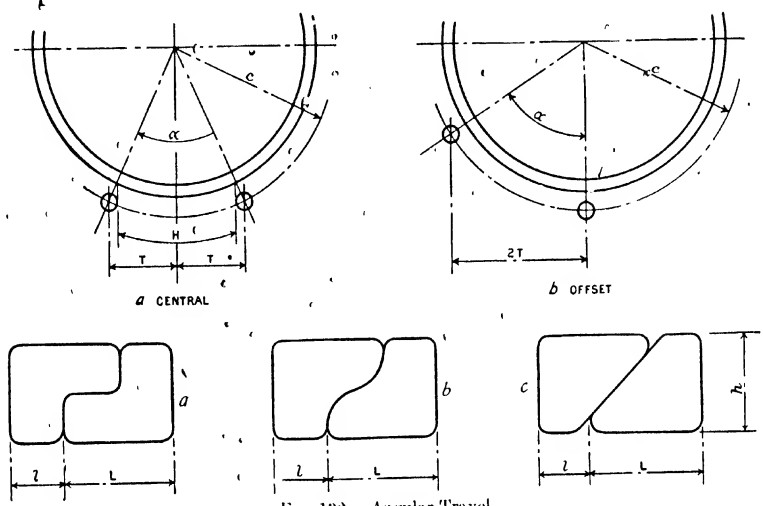

Fig. 132.—Angular Travel

settings are given in fig. 133, while fig. 134 illustrates in proportion several settings. It will be noticed that a "double purpose" port—that is, one which acts alternately as inlet and exhaust—is included in

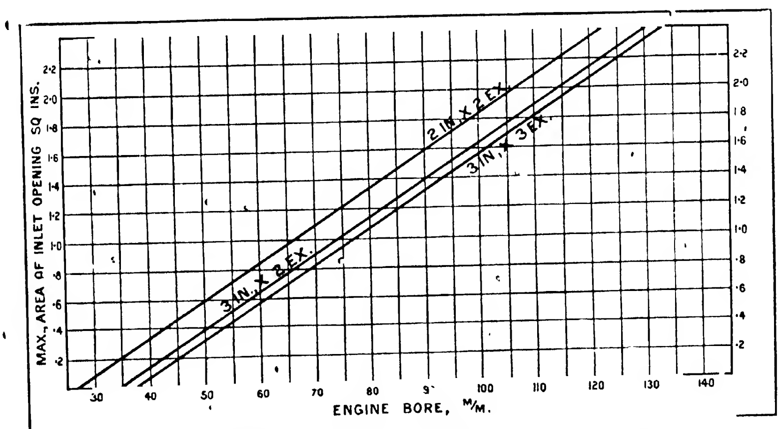

Fig. 133.—Maximum Inlet Valve Opening. Various Port Settings

each setting. This is advisable where maximum openings are desired; two single ports with a wall between would obviously use up more of the sleeve-valve circumference than a single "double purpose" port.

The sleeve valve is usually centrifugally cast of good quality grey iron, and for engines up to $2\frac{3}{4}$-in. diameter bore is made ·1 in. thick, while for engines of $4\frac{1}{4}$-in. diameter bore ·125 in. thick is quite satisfactory. D can readily be obtained by adding twice the sleeve-valve wall thickness to the cylinder bore diameter, while C, which should be kept as small as possible, is generally ·575D when the ball-and-socket type of drive is used.

The throw of sleeve crank T is obtained from the number of ports in the cylinder, as given by the setting adopted, and the dimensions D and C.

$$T = \frac{\pi D \times \cdot 575D}{D(2\ \text{No. of ports}) - 1} = \frac{1\cdot 8D}{(2\ \text{No. of ports}) - 1}.$$

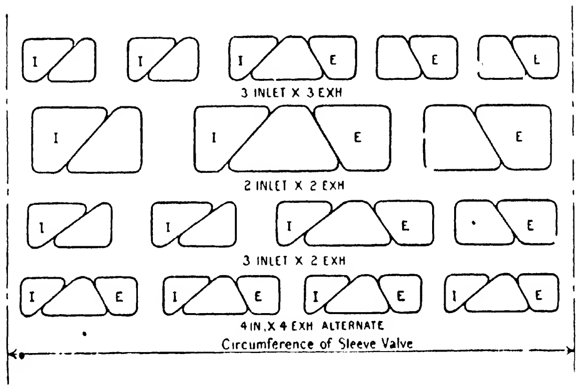

Fig. 134.—Various Port Settings

This gives the maximum throw possible with this type of actuation and may be reduced within limits according to the type of engine under consideration. Maximum throws for various settings and engine bore diameters are given graphically in fig. 135.

The valve timing V has now to be settled, and in common with poppet valves it varies according to the type of engine.

The actual calculation of ports will be best understood by taking an example. Assume a high-speed engine of 68-mm. bore, the desired maximum opening area of the inlet ports being 1·0 sq. in. approximately, so that we have

A = 2 inlet × 2 exhaust, for it will be seen from fig. 133 that this setting gives the required area.

D = 68 mm. + ·2 in. = say 2·9-in. diameter.

C = ·575 × 2·9 = 1·66 in., say 1·66 in.

$$T = \frac{1{\cdot}8 \times 2{\cdot}9}{(2 \times 4) - 1} = {\cdot}74 \text{ in., say } {\cdot}7 \text{ in.}$$

a = Angular travel of sleeve valve (see fig. 132).

$$\sin \tfrac{1}{2}a = \frac{T}{C} \text{ when T is central} = \frac{{\cdot}7}{1{\cdot}65} = {\cdot}42424 \text{ or } 25^\circ\ 6';$$

$$\therefore = 25^\circ\ 6' \times 2 = 50^\circ\ 12'.$$

When T is offset $\sin a = \dfrac{2T}{C}$.

Referring to fig. 136:

$$H = \text{Horizontal travel} = \frac{a\pi D}{360} = \frac{50^\circ\ 12' \times 3{\cdot}1416 \times 2{\cdot}9}{360} = 1{\cdot}275''.$$

L = Length of port = H − cover = 1·275″ − ·05″ = 1·225″.

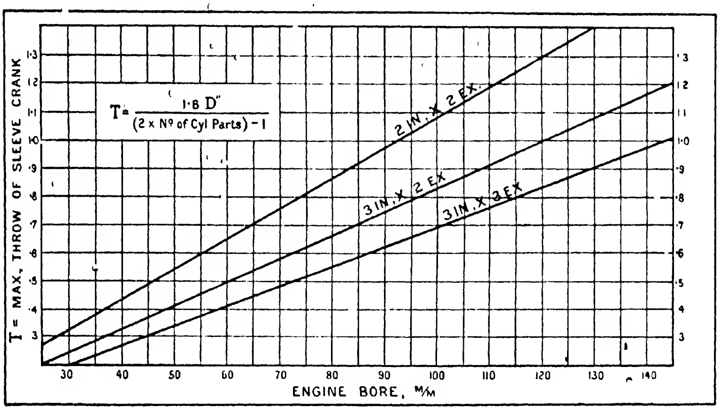

Fig. 135.—Maximum Sleeve-valve Crankthrow. Various Port Settings

Minimum cover = ·04″. This is usually arranged so as to bring L to an even figure.

W = Minimum space between ports (this does not apply to alternate ports as shown in setting *d*, fig. 134) = H + cover = 1·275″ + ·05″ = 1·325″.

h = Height of ports = T + (T sin β) = ·7 + (·7 sin 22½°) = ·97″.

4 ports at 1·225″ long = 4·9″
2 spaces at 1·325 „ = 2·65″
2 spaces at ·775 „ = 1·55″

Circumference of sleeve valve = 9·1″.

$$li = \text{Inlet port tail} = \frac{H}{2} - \left(T \sin \delta \frac{{\cdot}5D}{C}\right),$$

$$= {\cdot}6375 - \left({\cdot}7 \sin 7\tfrac{1}{2}^\circ \times \frac{1{\cdot}45}{1{\cdot}65}\right) = {\cdot}6375 - {\cdot}08 = {\cdot}5575''.$$

$$le = \text{Exh. port tail} = \frac{H}{2} + \left(T \sin \nu \frac{\cdot 5D}{C}\right)$$

$$= \cdot 6375 + \left(\cdot 7 \sin 15^\circ \times \frac{1\cdot 45}{1\cdot 65}\right) = \cdot 6375 + \cdot 1595 = \cdot 797''.$$

With straight-sided ports the flank angle can be solved as follows (see fig. 136) :—

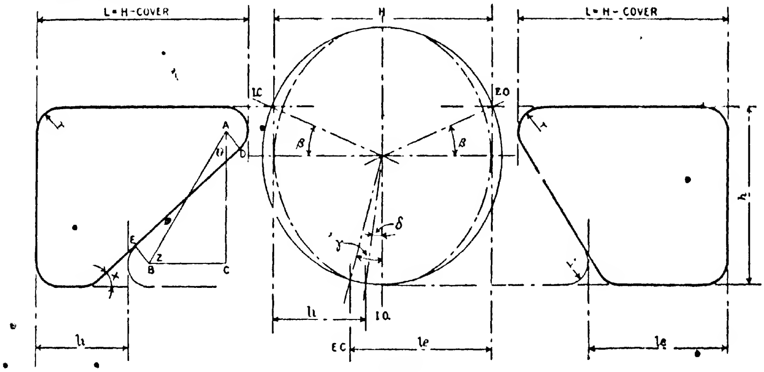

Fig. 136

$AC = h - 2r,$ $\tan \theta = \frac{BC}{AC}.$

r = corner radius usually 1/8″, $AB = \frac{BC}{\sin \theta}.$

AD and EB $= r,$ $\sin \phi = \frac{r}{\frac{AB}{2}}.$

$BC = L - (l + 2r),$ $z = 90^\circ - \phi.$

Flank angle $= X = z - \phi.$

In fig. 137 the crankshaft timing diagram for the example worked

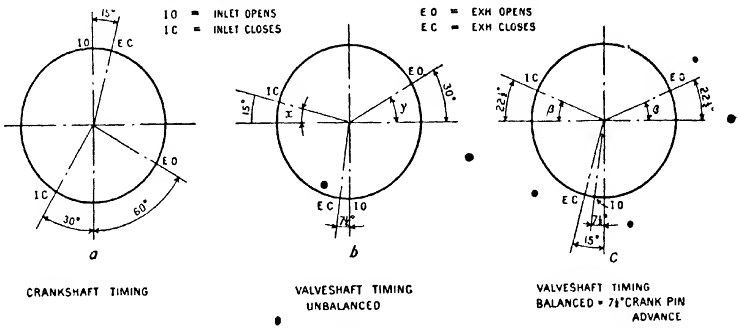

Fig. 137.—Timing Diagrams

is shown at a, while the same timing transferred to the valveshaft is indicated at b, the crankshaft being at TDC, while the valveshaft crankpin is at BDC. It is obvious that with this valveshaft setting

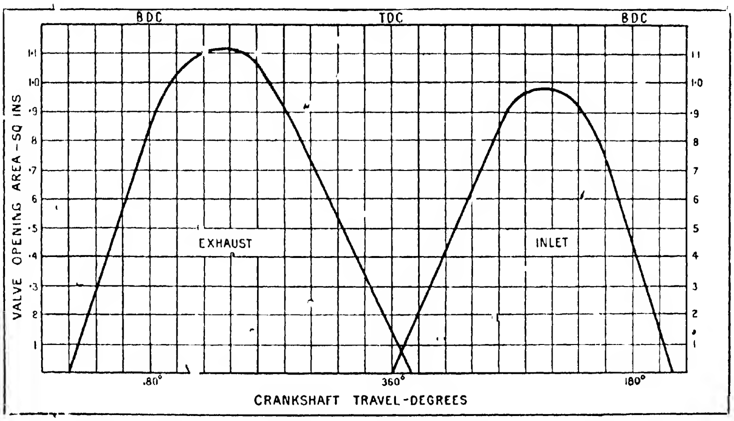

Fig. 138. Valve-opening Diagram. 68-mm. Bore Engine; C = 1·65″, T = ·7″

a relatively large opening to exhaust would be obtained owing to the greater height of exhaust port. To overcome this, and in order to make the machining of ports a simpler operation, inlet and exhaust

Fig. 139.—Four-cylinder Burt Single-sleeve Valve Engine

ports are made of equal height. This is made possible by the setting of valveshaft crankpin in advance of its BDC and in relation to the crankshaft until the angles x and y are equal. The amount of offset

is found by $\frac{x+y}{4}$, and the result for the example taken is shown in fig. 137 at c: the corresponding port opening diagram being as shown in fig. 138.

Fig. 139 shows an example of a small four-cylinder motor-car engine of 68-mm. bore and 103 mm. in which single-sleeve valves actuated by the ball-and-socket mechanism are employed; while

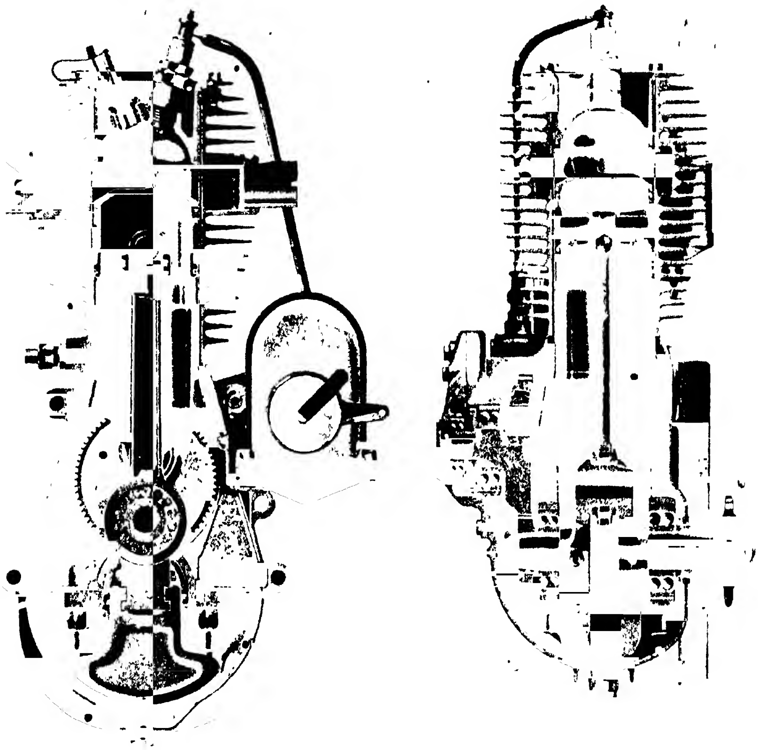

Fig. 140 Fig. 141

Barr & Stroud Engine with Burt Single valve

figs. 140 and 141 show a remarkably neat design of motor-cycle engine built by Messrs. Barr & Stroud, and incorporating the same features.

Racing Cars.—The practice of motor-car and cycle racing has been one of the most valuable stimulants to the design of efficient internal-combustion engines, for the racing engine operates under conditions of severity such as are met with in no other field, with

the result that weaknesses which would develop in the course of years under ordinary conditions of service are shown up in as many minutes. The rapid progress which the high-speed internal-combustion engine has made during recent years is due to the stimulus of motor racing, and that to an extent which few people fully realize.

In the production of a racing engine the designer has full liberty to employ every means known to him to obtain the highest possible power output regardless of any other consideration except that during more recent years it has become the practice to limit the cylinder capacity of racing engines, a restriction which has proved of undoubted benefit.

There is a popular impression that because the racing engine no longer resembles the actual article used in touring cars its value, from an educational point of view, has been lost; this, however, is a sheer fallacy: the racing engine operates on the same cycle and under the same conditions, except that they are much more severe, as the ordinary touring-car engine, and the lessons learnt from its behaviour are just as applicable to the intelligent designer as though the engines were identical.

From an educational point of view it is probably desirable that the racing engine should differ from the touring model, for by its difference—

(1) Higher speeds and therefore more strenuous conditions of test are obtained.

(2) The racing engine of to-day is providing lessons for the future also, and not only for the immediate present.

Again, there is a popular but wholly mistaken belief that the racing motor-car engine, though powerful, is not "efficient," and that since fuel economy does not enter into consideration such engines teach us nothing about this important question. For an engine to be powerful it must be efficient in every respect—that is to say, it must convert the highest possible percentage of the heat energy available from the combustion of every pound of air into useful work at the flywheel. If, as may sometimes be the case, the air is supersaturated with fuel, that is the carburettor's not the engine's fault, for with good carburation the racing engine will show the highest possible thermal efficiency reckoned on the fuel also.

The engine illustrated in figs. 142-149 is one of several constructed by Messrs. Vauxhall Motors Ltd. for their racing cars for the 1922 season. It is of three-litres capacity, and develops, the author believes,

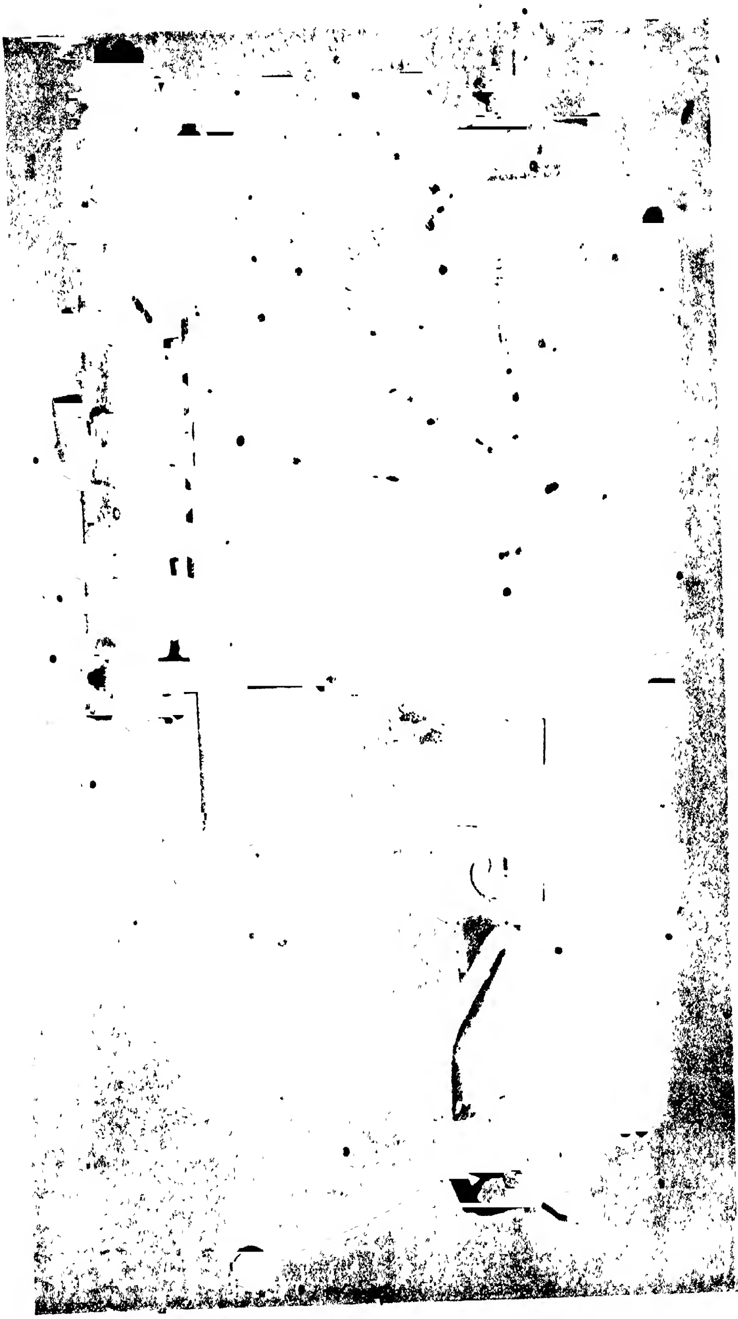

Fig 142.—Vauxhall Three litre Racing Engine

Fig. 143 — Vauxhall Three-litre Racing Engine

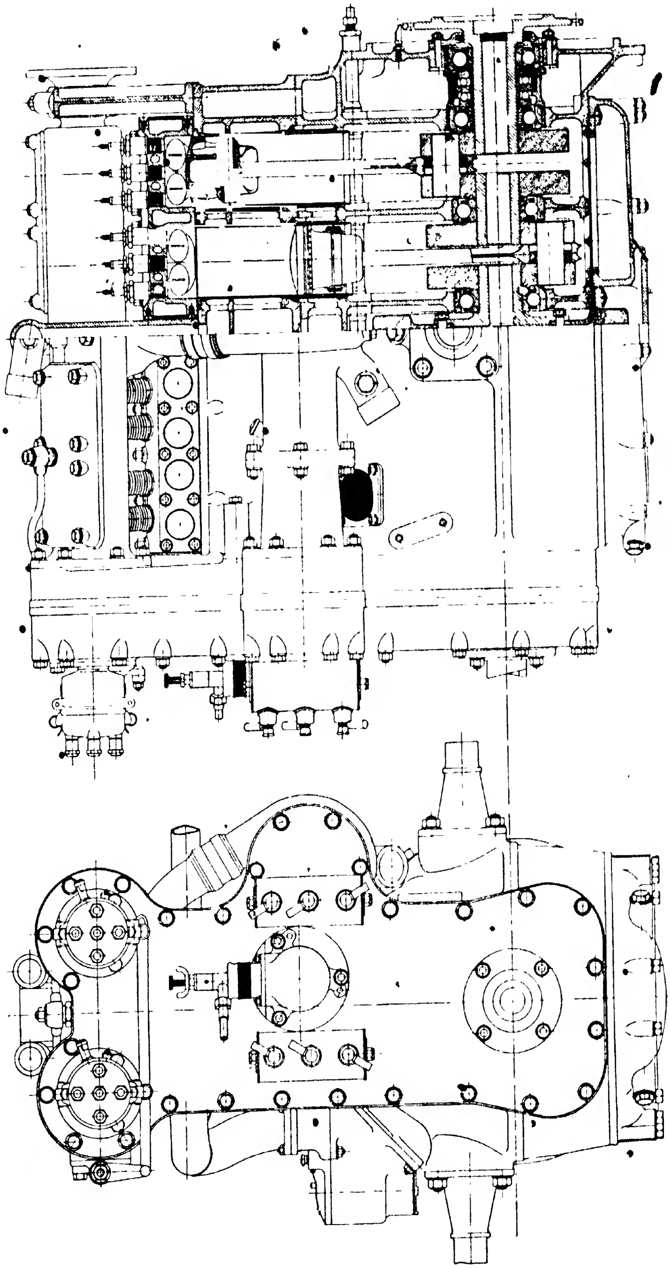

Fig. 144 —Vauxhall Three-litre Racing Engine

the highest power output ever yet obtained from an engine of this size. The main features aimed at in the design of this engine were :

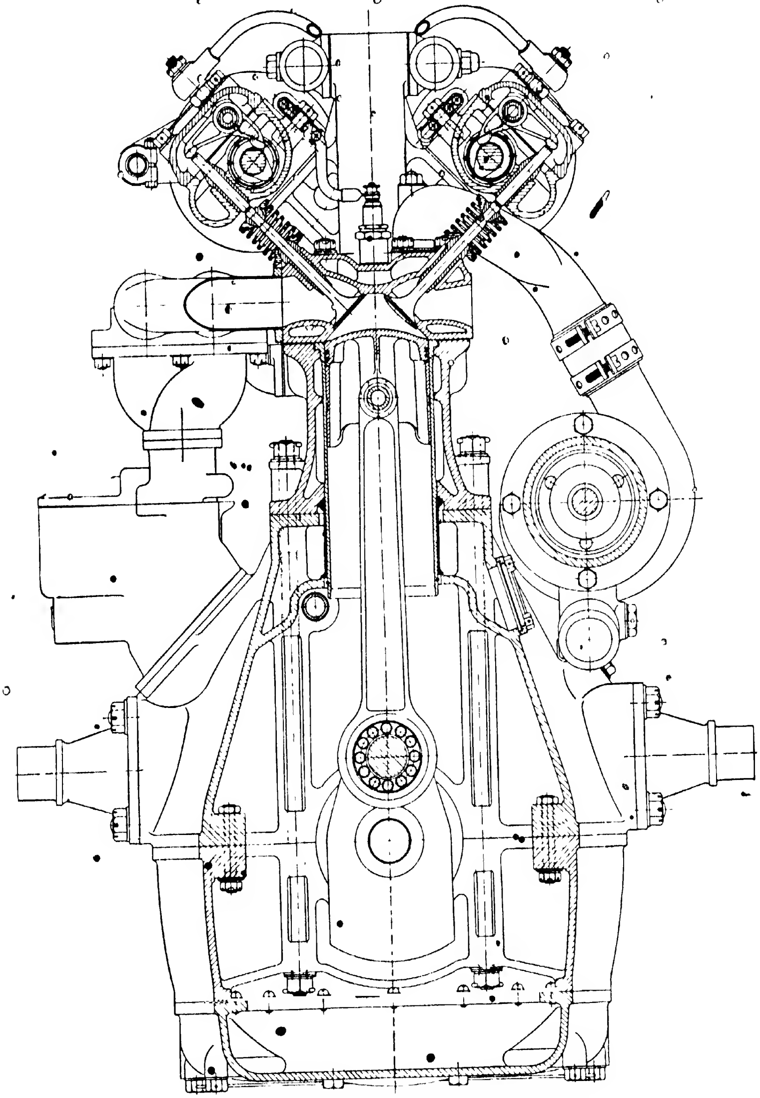

Fig. 145.—Vauxhall Three litre Racing Engine

(1) To obtain the maximum possible thermal efficiency, with a view to getting the utmost possible power output from the available air.

(2) To ensure the maximum of structural rigidity.

(3) To avoid crankshaft torsional vibration at any speed of which the engine was capable.

(4) To obtain the highest possible mechanical efficiency.

(5) To obtain a high volumetric efficiency.

(6) To provide a form of connecting-rod big-end bearing which

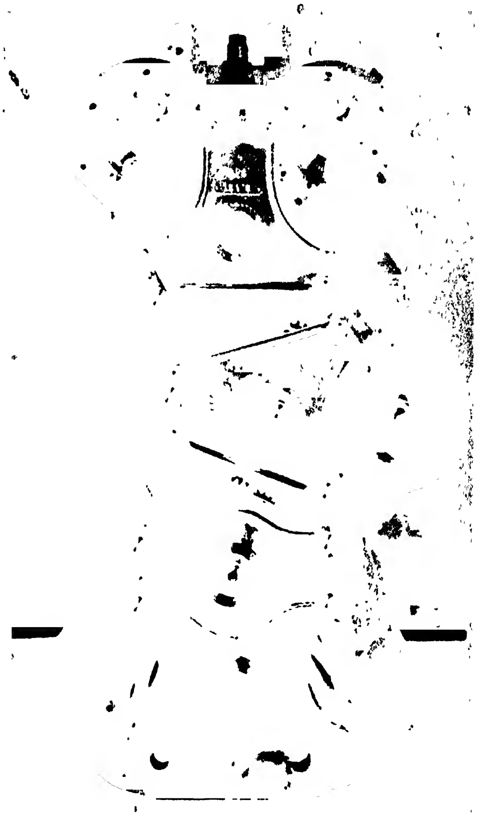

Fig. 146.—Vauxhall Three-litre Racing Engine: Auxiliary Drive

should be capable of withstanding continuous running at a mean speed of well over 4000 R.P.M.

The steps taken to meet these conditions were :—

(1) In order to obtain the highest possible thermal efficiency the combustion chamber was made of the shallow pent-roof type with the sparking fitted centrally in the cylinder head. The maximum

distance from the sparking plug points to the farthest point in the combustion chamber is only 1·9 inches. In addition to the central plug, provision was made for the fitting of two other plugs, one on either side, to be operated synchronously from a single low-tension contact breaker. These additional plugs were intended rather as a standby in case of failure of the central plug; they were, in fact, never used.

(2) In order to ensure the maximum of structural rigidity the crankcase is made as deep as possible and of a barrel shape, with the maximum cross-section at the centre; the cylinder water-jackets were cast in one piece and rigidly attached to form an additional

Fig. 147.—Vauxhall Three-litre Racing Engine: Cylinder Heads and Valves

girder, while through bolts extend from the cylinder block to the very bottom of the crankcase, thus forming a structure of extreme rigidity both as regards torsion and bending.

(3) With a view to eliminating torsional vibration, the flywheel is mounted in the centre of the crankshaft so that the maximum length subject to torsion is reduced to about 8 inches. The shaft is, in fact, made in the form of two entirely separate two-throw cranks, each provided with flanges between which the flywheel is bolted. This arrangement, although very unorthodox, proved most successful, no trace of torsional vibration being observed at any speed at which the engine could be run.

(4) With a view to obtaining the highest possible mechanical

efficiency, pistons of the slipper type are employed and the cylinder liners are maintained at a high temperature in order to lower the viscosity of the lubricant adhering to them. This latter is accom-

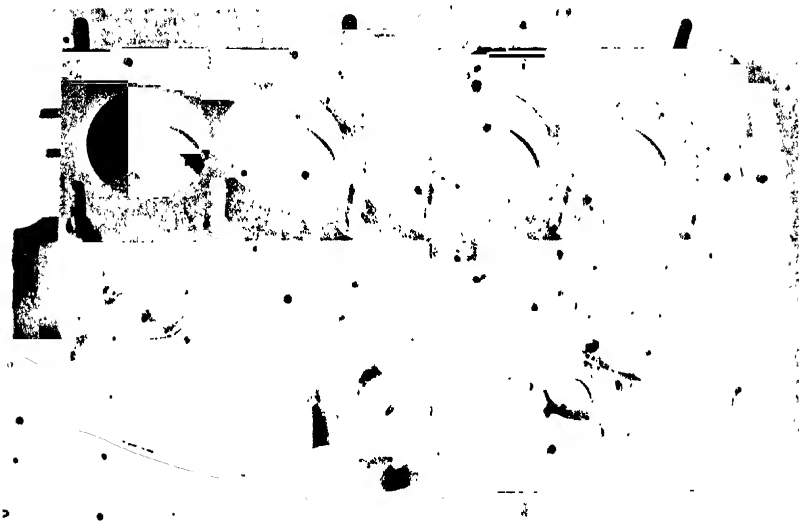

Fig. 148.—Vauxhall Three-litre Racing Engine: Crankcase

plished by isolating the lower part of the liners from the main water circulation, so that the cooling water surrounding them is left practically stagnant. For the rest, the use of ball and roller bearings wherever possible contributed to reducing the friction losses to the

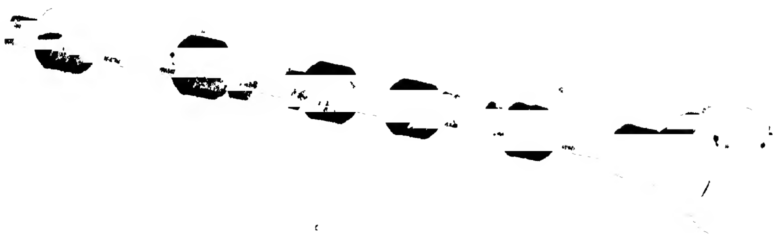

Fig. 149.—Vauxhall Three-litre Racing Engine: one of the Camshafts

lowest possible limit and at the same time obviated the necessity for any form of oil cooling—always a troublesome problem.

(5) In order to obtain a high volumetric efficiency the induction system was divided so as to avoid any overlapping of the suction strokes. The central pair of cylinders were fed from one carburettor

and the outer pair from a second and entirely independent carburettor. In this manner use could be made of the kinetic energy of the gases flowing in the induction pipes thoroughly to fill the cylinders without any risk of one cylinder robbing another, as must inevitably occur when all four cylinders draw from any one induction manifold, owing to the overlapping of the opening period of the inlet valves.

The engine has four cylinders, each of 85 mm. bore and 132 mm. stroke, and was designed with a view to running continuously at from 4000 to 4500 R.P.M. with short periods up to 5000 R.P.M. In order both to provide structural rigidity and to enclose the central flywheel the crank chamber is of barrel shape and is mounted in the chassis by trunnions attached to the sides. The cylinder block consists of an aluminium casting forming the water-jacket into which loose steel liners are fitted, with rubber rings to ensure water-tightness. The cylinder heads are cast in pairs in hard bronze and call for no particular comment. The valves, of which there are four in number to each cylinder, are comparatively small with a high lift, and the valve gear generally is designed to operate at a speed of 5000 R.P.M. The inlet valves are heavily masked, and by this means it is possible to employ a comparatively low rate of acceleration and therefore to use very light and lightly stressed valve springs. The two camshafts are carried in aluminium housings supported from the main cylinder block at their centre and at either end. They run in plain bearings with cast-iron floating bushes. The cams themselves are of very small diameter in order to reduce to the minimum the rubbing velocity; they are of the plain tangent flank form. The cam followers are of the plain curved slipper type, with a short straight push rod interposed between the follower and the valve itself. The camshafts are driven by means of a chain of spur gears, the intermediate pinions of which are carried in separate spider housings to permit of the meshing being correctly adjusted.

(6) Experience with bearings, both in actual engines and under separate tests, had shown that, even under the most favourable conditions as to lubrication, plain white-metal lined bearings could not be relied upon for the connecting-rod big-ends, because no matter how much oil be circulated through the bearings, nor how thoroughly it be cooled, there was little hope of getting rid of the heat generated by friction at a rate sufficient to keep the temperature of the bearing material within safe limits. Further, in order to reduce vibration and ensure structural rigidity, it was essential

to keep the cylinder centres as close together as possible, and this limited the permissible width of bearing surface both on the crank-pins and main crankshaft journals.

In view of these considerations it was apparent that if a continuous mean speed of over 4000 R.P.M. was to be maintained some form of crankpin bearing other than a plain white-metal lining would have to be employed. The choice lay between (*a*) the use of a floating bush between the crankpin and connecting-rod, under forced lubrication, and (*b*) a roller bearing. Both these would necessitate the use of some form of built-up crankshaft with case-hardened crankpins, since neither the floating bush nor the roller race could be split, nor could the weight of a split big-end bearing be tolerated. Of the two alternatives the floating bush would require continuous lubrication under pressure, while the roller bearing could be used with splash lubrication. Since for the same and other additional reasons, it was essential to employ ball or roller bearings for the main journals, the provision of continuous lubrication under pressure to the crankpins became a very difficult problem, and it was decided therefore to adopt the second alternative and employ roller bearings. The method of building up the crankshaft was another problem, and after much consideration it was decided to employ a completely built-up crank, consisting of plain parallel pins on to which the crank-cheeks were shrunk as in marine and large gas-engine practice.

The crankshafts throughout were made from plain mild steel with the pins case-hardened. For the connecting-rod big-end bearing, it was decided to use a double row of short rollers located in a one-piece bronze cage, while the hardened eye of the connecting-rod itself formed the outer race. This is further stiffened by means of two circular webs. Like the crankshafts, the connecting-rods are of plain low-carbon case-hardened steel.

Lubrication.—Two oil pumps of the oscillating valveless plunger type are provided, both of which are operated from one of the idle wheels of the gear train. One pump draws oil from the oil sump and delivers it to an oil gallery running the full length of the crank-case, provided with four jets playing oil on to each of the crank-throws. The second pump delivers oil under a pressure of about 25 lb. per square inch to the camshafts, the oil being distributed through the hollow fulcrum pins of the valve rockers; from the camshaft casings, the oil drains back by gravity to the crankcase.

Cooling.—The cooling water is circulated by means of a centri-

fugal pump running at one-half engine speed. From the pump the water passes around the upper end of the cylinder liners, the lower parts of which are partitioned off in order to maintain the water more or less stagnant, and so to permit of the bearing portion of the liners attaining rapidly a fairly high temperature. From the upper deck of the cylinder block the water passes to the cylinder heads in parallel, and so returns to the radiator. In order both to cool the crankcase and slightly to heat the air on its way to the carburettors, the air supply to the engine is drawn through the upper part of the crankcase and around the exposed lower ends of the cylinder liners.

The compression ratio used is 5·8 : 1. It had originally been proposed to employ a much higher compression ratio, and to run on a special fuel mixture, but owing to the difficulty of providing an efficient form of combustion chamber with any higher compression ratio, and at the same time avoid any risk of the pistons striking the valves should these accidentally stick in the full open position, it was considered safer to employ a lower ratio, at which, owing to the short-flame travel from the sparking plug, ordinary good quality petrol can be used without detonation.

In general, though the engine is designed throughout to run at very high speeds, it contains no extremes either of design or material. The whole of the crankshaft, the connecting-rods, and the gudgeon pins are of straight low-carbon mild steel. Neither the connecting-rods nor the pistons are particularly light. The cams are of the plain tangent flank form, free from any concave surfaces, the acceleration of the valve gear is low, and the valve springs are very lightly stressed. In short, the engines were designed throughout with a view to reliability both in manufacture and in running, and an ample margin of safety provided for.

The performance of one of these engines which underwent prolonged testing on the test-bench is shown in fig. 150, from which it will be seen that a maximum of 129 B.H.P. is reached at a speed of 4500 R.P.M., the brake and indicated mean pressures at this speed being 124 and 159 lb. per square inch respectively, and the mechanical efficiency 78 per cent. It will be observed also that the highest indicated mean pressure was obtained at a speed of about 3700 R.P.M., showing that the combination of induction pipe design and valve setting was such as to give maximum over-all efficiency at this speed. With a combustion efficiency of 34·75 per cent this corresponds to a volumetric efficiency at N.T.P. of 80·3 per cent, a figure which is in very close agreement with that obtained from the author's

variable compression engine under similar conditions as to temperature, and at a speed of 1750 R.P.M., which is the most efficient speed for this particular engine. This figure also is in very close agreement with readings taken of the compression pressure when motoring, which at 4000 R.P.M. was found to be 139 lb. per square inch, indicating that the cylinders were filled up to very nearly full atmospheric pressure at this speed. The mechanical efficiency was arrived at from a very large number of tests by Morse's method on pairs of cylinders separately, on individual cylinders, and by motoring tests. The three methods showed exceptionally close agreement over the whole range of speed.

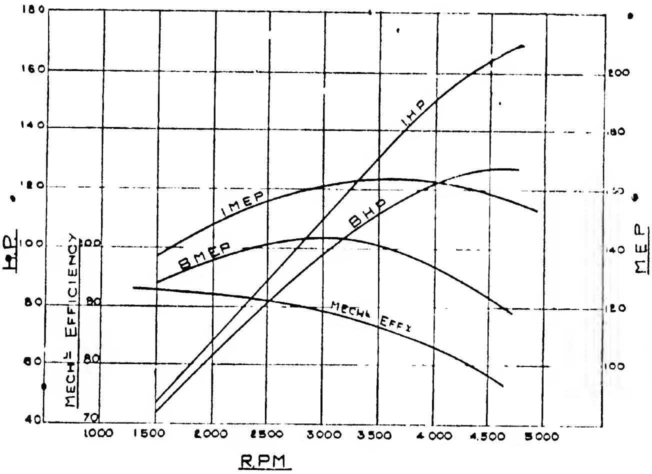

Fig. 150 —Vauxhall Three litre Racing Engine

The performance of the engines is almost exactly what might be anticipated from an analysis of the general design based on the data given in the preceding chapters, and, since it conforms so closely, it may be of interest and perhaps of some use to engine designers to recapitulate such data in so far as it applies to these particular engines. It may also, the author hopes, help to dispel the still prevalent superstition that some mystery enshrouds the performance of racing motor-car engines, whose behaviour is, in fact, perfectly normal in every respect.

The leading dimensions of these engines are as follows :—

Bore	85 mm. = 3·34 inches
Stroke	132 mm. = 5·2 ,,
Compression	5·8 : 1.

Area of piston	8·75 square inches.
Swept volume of cylinder	45·5 cubic inches.
Weight of reciprocating parts (per cylinder)	1·7 lb.
„ „ „ (per sq. in. of piston area)	0·195 lb. per sq. in.
Number of valves	4—2 inlet, 2 exhaust.
Diameter of valve ports (inlet)	=1·34 inches.
„ „ (exhaust)	=1·30 „
Lift of all valves	=0·354 „
Effective area through inlet valves	=2·55 sq. in.
(Inlet valves masked 0·050 inches of travel.)	
Ratio piston area to effective inlet port area	=3·44 : 1.

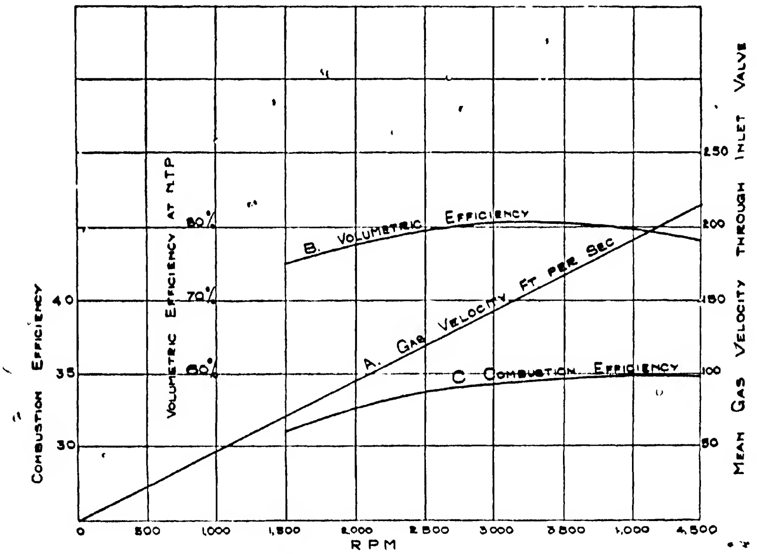

Fig. 151.—Vauxhall Three-litre Racing Engine

The chart, fig. 151, curve A, shows the mean gas velocity through the inlet valves at speeds varying up to 4500 R.P.M., and curve B shows the volumetric efficiency which corresponds with these mean gas velocities allowing for the latent heat of evaporation of the fuel, assuming that the minimum of preheating were used and that there were no undue wiredrawing in the induction pipe or carburettors, both of which conditions apply in this case. Curve B is arrived at by calculations such as those given in Chapter II, by deduction from various test results, and finally by direct air measurements taken on various engines with similar combustion chambers and similar gas velocities through the inlet valves. The falling off

in volumetric efficiency at the lower speeds is due merely to the late closing of the inlet valve, which results in the rejection of some of the combustible mixture during the early part of the compression stroke; while the falling off again at very high speeds is due to wiredrawing or insufficient valve area. With a combustion chamber of the form used giving the maximum of turbulence, and with central ignition, the efficiency reckoned on the air consumption will, at high speeds, be approximately 69 per cent of the air-cycle efficiency for the compression ratio (*vide* Chapter IV).

The air-cycle efficiency corresponding to a compression ratio of 5·8 : 1 is 50·5 per cent, so that the combustion efficiency will be $\frac{69}{100} \times \frac{50 \cdot 5}{100} = 34 \cdot 8$ per cent at the highest speeds, when turbulence is at a maximum and direct heat loss at a minimum. As the speed is reduced turbulence becomes less, owing to the lower entering gas velocity, and the direct heat loss, though of comparatively small consequence, will also increase, with the result that combustion efficiency may be expected to vary approximately as the curve C, fig. 151. The values given in this curve are taken from those found on the author's variable compression engine under almost exactly similar conditions.

From the curves B and C and the known heat of combustion per standard cubic inch of mixture, as given in Chapters I and II, the indicated mean effective pressure can be arrived at directly by multiplying together the volumetric efficiency, the heat of combustion per standard cubic inch (47·8 ft.-lb. in this instance) × 12 × the combustion efficiency; thus at 3000 R.P.M. the indicated mean pressure will be 0·807 × 47·8 × 12 × 0·343 = 159 lb. per square inch. Similarly at speeds from 1500 to 4500 we find that the theoretical indicated mean effective pressure should be—

R.P.M.	Volumetric Efficiency per cent.	Combustion Efficiency per cent.	Indicated mean Pressure lb. per sq. in.	Observed I.M.E.P. lb. per sq. in.
1500	75·0	31·0	133·4	138·5
2000	77·6	32·6	145·1	148·0
2500	79·7	33·9	155·0	156·0
3000	80·7	34·3	159·0	161·0
3500	80·7	34·6	160·4	162·5
4000	79·9	34·7	159·2	162·2
4500	78·5	34·8	156·9	159·0
5000	77·0	34·8	153·7	...

From the above it will be seen that the agreement between the estimated and the observed figures is so close as to prove that the engine is behaving normally in every respect; it is, in fact, so close as to indicate a certain share of coincidence, for neither calculation nor measurement could be exact to within closer limits than 1 per cent, leaving scope for a variation of 3 lb. per square inch between the observed and the calculated figures, assuming that all the premises on which the latter were based were strictly accurate. No tests were made above 4500 R.P.M. or below 1500.

In the first volume of this book certain empirical formulæ, based on an accumulation of experimental results, were given for determining the mechanical efficiency of an engine. These formulæ were arrived at from a large number of experiments mostly carried out on quite slow-running engines, but later experiments indicate that they are applicable also to high-speed engines with, perhaps, certain small reservations.

The losses in any internal-combustion engine may be divided up into—

(1) Piston friction.
(2) Fluid pumping losses.
(3) Bearing friction and auxiliary drives.

Of these piston friction constitutes always by far the largest proportion, and all more recent tests appear to indicate that for a piston of more or less normal design and proportions, and for normal conditions as to lubrication and jacket temperature, the piston friction in terms of lb. per square inch on the piston head may be arrived at with a fair degree of approximation by the empirical formula—

$$\left(\frac{\dfrac{\text{Mean fluid pressure including compression}}{4} + \dfrac{2\text{ Mean inertia pressure}}{3}}{10}\right) + 2\cdot 0.$$

From such a formula we find that the piston friction expressed in terms of mean pressure on the piston at every fourth stroke (i.e. expressed on equal terms to the useful mean pressure) should be—

Speed R.P.M.	Piston Friction. (lb. per sq. in. M.E.P.)	Speed R.P.M.	Piston Friction. (lb. per sq. in. M.E.P.)
1500	7·7	3500	13·1
2000	8·6	4000	15·1
2500	10·0	4500	17·3
3000	11·4		

The fluid pumping losses are dependent, assuming a normal valve setting, as in this case, and no other serious obstruction to the flow of gas into or out of the cylinders, upon the mean gas velocity through the valves and more particularly through the inlet valves. In volume I, fig. 23, a curve is given showing the observed fluid pumping losses at different gas velocities: from this curve, and that of gas velocity given in fig. 151, we find that the fluid pumping losses in this particular engine again expressed in terms of mean pressure should amount to—

Speed R.P.M.	Fluid Pumping Losses. (lb. per sq. in.)	Speed R.P.M.	Fluid Pumping Losses. (lb. per sq. in.)
1500	2·0	3500	5·4
2000	2·3	4000	6·9
2500	2·9	4500	8·6
3000	4·0		

Finally, there remains the friction of the bearings and the power absorbed by the auxiliary drives: these latter consist of a long train of gears to operate the overhead camshafts, the camshafts themselves in plain floating bearings, a large water circulating pump, two oil pumps, a small air-compressor for fuel supply, and the ignition gear. No direct measurement was taken of the power absorbed by these auxiliaries, and in the absence of actual data we must fall back on analogy from tests on other engines more or less similarly equipped. From such an analogy it may be assumed that the loss due to all these sources will range in more or less a straight line from the equivalent of 3 lb. per square inch at 1500 R.P.M. to about 5 lb. at 4500 R.P.M.

From the above it will be seen that the total of fluid and frictional losses may be estimated at—

Speed R.P.M.	Total Losses. (lb. per sq. in. M.E.P.) estimated	Observed Losses.
1500	12·7	10·9
2000	14·2	12·6
2500	16·5	15·2
3000	19·3	18·6
3500	22·7	22·7
4000	26·6	28·8
4500	30·9	35·0

Fig. 152 shows in full lines the estimated friction losses as arrived at in the above tables, and in dotted line, the observed losses as

arrived at by motoring and by tests with the ignition cut off from various cylinders. It will be observed that while the general slope of the curve of mechanical and other losses does not agree very closely with the estimated curve, yet the mean value throughout the whole range of speed is in very fair agreement.

A number of readings of fuel consumption using the petrol referred to—as sample A in Chapter I—were taken at different speeds and loads, using in all cases an economical carburettor setting: that is to say, about 10 per cent weak as against the 20 per cent rich mixture employed for the attainment of the utmost possible power output. The fuel consumption readings therefore were taken with

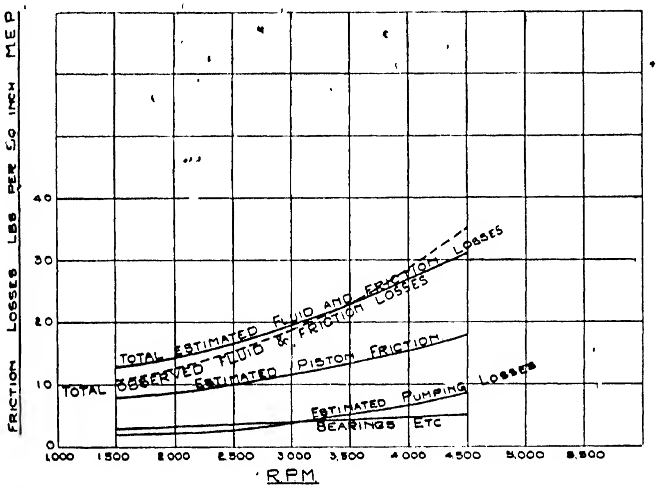

Fig. 152.—Estimated and observed Fluid and Friction Losses

the carburettor so adjusted as to reduce the maximum power by about 6 per cent. These tests gave the results shown in figs. 46 and 47, in terms of pints per hour per indicated and per brake horsepower. This particular petrol has a corrected calorific value (including the latent heat of evaporation of the liquid) of 18150 B.T.U.s per pint.

It will be seen that at 3000 R.P.M. the fuel consumption on full load is only 0·45 pint per B.H.P. and 0·395 pint per I.H.P. hour, corresponding to a brake thermal efficiency of 31·2 per cent and an indicated thermal efficiency of 35·4 per cent, a figure actually slightly in excess of the computed combustion efficiency, while at 66 per cent full load torque the observed fuel consumption at about

this speed was only 0·48 pint per B.H.P. hour, a figure unequalled even at full load by any touring-car engine. These figures should

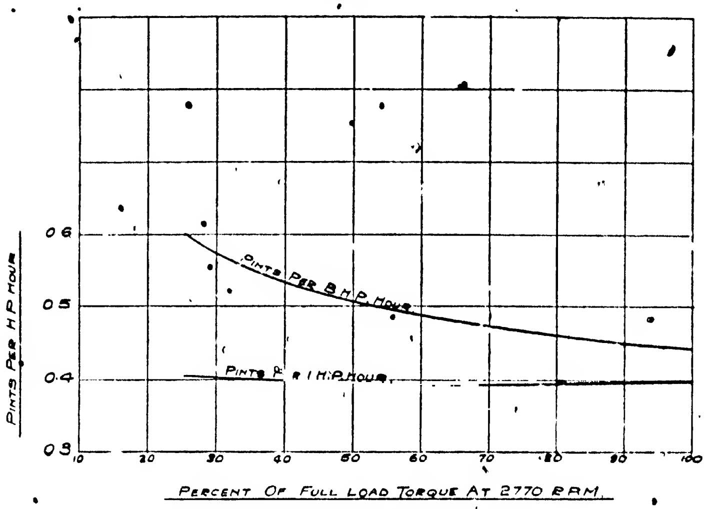

Fig. 153.—Three-litre Vauxhall Racing Engine
Curve showing Consumption on Throttle at 2770 R.P.M., and most Economical Mixture Strength. Fuel, Petrol. Oil, "Shell," L.R.O.

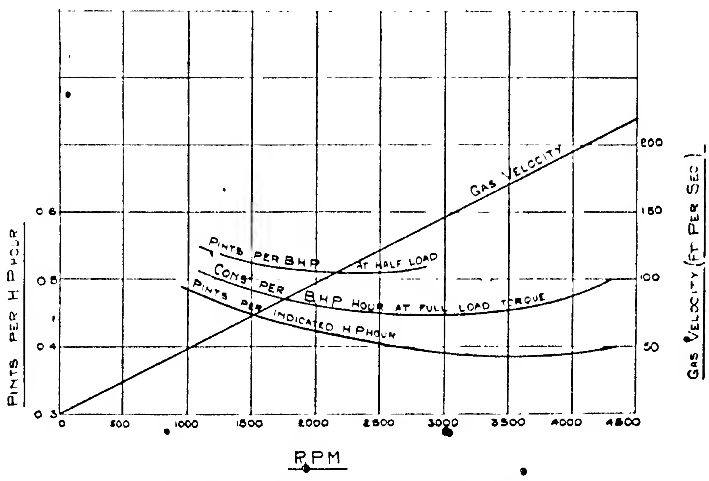

Fig. 154.—Three-litre Vauxhall Racing Engine
Curve showing Variation in Fuel Consumption with Speed. Most Economical Mixture in all cases. Fuel, Petrol. Oil, "Shell," L.R.O.

go far to dispel the theory that a racing-car engine is essentially extravagant in fuel. The readings of fuel consumption when the

load was reduced by throttling down to one-third full load torque are rather striking.

Thus at 2770 R.P.M. the fuel consumption on petrol was found to be—

B.H.P.	Fuel Consumption. (Pints per B.H.P. hour.)
80	0·455
70	0·465
60	0·480
50	0·505
40	0·535
30	0·565
20	0·610

At 1950 R.P.M. the fuel consumption was—

B.H.P.	Fuel Consumption. (Pints per B.H.P. hour.)
60	0·470
50	0·480
40	0·495
30	0·525
20	0·570
15	0·605

It will be noted that when developing 20 B.H.P. at 1950 the gross consumption is only $20 \times 0{\cdot}570 = 11{\cdot}4$ pints per hour. With the gear ratio used 1950 R.P.M. corresponds to a road speed of 48·5 M.P.H., a speed which calls for an expenditure of just about 20 B.H.P. reckoned at the engine shaft, so that the consumption in miles per gallon with an economical carburettor setting even at this high mean speed should be $48{\cdot}5 \times \frac{8}{11{\cdot}4}$ = about 33·9 miles per gallon.

A brake thermal efficiency of 31·2 per cent is, the author believes, the highest ever yet achieved by any engine running on petrol. Incidentally it is worthy of note that the indicated thermal efficiency reckoned from the fuel consumption, when using a weak mixture, corresponds very closely with the calculated combustion efficiency, showing that there can be practically no loss of unburnt fuel by irregular distribution or indeed from any other cause. It is interesting also to note that the fuel consumption per I.H.P. hour is exactly the same as that obtained in the single-cylinder variable compression engine (described in Chapter I) when running on the same fuel, at the same compression ratio, and at the same gas velocity through

Fig. 155 —Vauxhall Three litre Racing-car

the valves; but the brake thermal efficiency of the racing engine is considerably greater owing to the higher mechanical efficiency of the latter.

The author has dealt at considerable length with this particular engine, because the results obtained from it serve admirably to emphasize that a racing-car engine is nothing more or less than a highly efficient internal-combustion engine designed throughout on purely scientific lines, and whose behaviour is, from a thermodynamic point of view, perfectly normal in every respect.

Fig. 155 shows a photograph of one of the racing-cars fitted with the engine described above. Fully equipped, as shown in the photograph, the car weighs with driver and mechanic 2700 lb.; in this condition and with a gear ratio giving 25 miles per hour at 1000 R.P.M. it is capable of a maximum speed on the level of 115 miles per hour, corresponding to an engine speed exclusive of wheel slip of 4600 R.P.M., the actual engine speed being probably about 4800 R.P.M.

CHAPTER XI

AERO-ENGINES

Of all the applications of the Internal-combustion Engine, it is to aircraft in particular that high efficiency in its widest sense is most essential. The aero-engine must be efficient not only in relation to the fuel it consumes, but in every possible respect, including the material of which it is constructed, and it is therefore primarily to the aero-engine that most of the considerations in the preceding chapters have been directed.

Although it is only a very few years since the first power-driven aeroplane succeeded in leaving the ground, yet in this very short space of time the aero-engine has passed through several phases of its development.

In its earliest stages of development the one controlling factor was weight, then, as the aeroplane improved and longer flights were contemplated, the weight, not of the engine itself, but with fuel, oil, etc., for a protracted flight became the primary consideration, and extreme lightness of the engine alone, began to give way to some extent before economy in fuel and oil consumption, and reliability.

The Great War broke out during a very early stage in the development of aircraft, but the importance of the airship and aeroplane for military and naval purposes became so obvious that development was stimulated to an extent which has probably never before occurred in any branch of engineering.

The beginning of the war found Germany, alone of all the nations concerned, with any form of considered policy in regard to the type and line of development to be pursued. France possessed comparatively, a very large number of aeroplanes propelled by every conceivable type of engine, including air-cooled, water-cooled, fixed radial and rotating radial, four-, six-, eight-, and twelve-cylinder stationary types, in fact a heterogeneous collection representing examples of every conceivable type, but apparently without any

policy as to which types to perpetuate for immediate military purposes. Our own country possessed very few aeroplanes at all, and still less experience. Such few engines as we did possess were propelled for the most part by a miscellaneous collection of French and German engines, a few by the R.A.F. Vee-type air-cooled engines, and one or two other more or less experimental English-designed engines of the straight-line six or Vee type. America, by watching the trend of events for over two years as a neutral, had ample opportunity to frame a policy, and decided, on entering the war, on the development of a twelve-cylinder Vee-type engine, embodying the proved features of the best Vee-type engines in use at the time. Though eventually a very satisfactory engine, its development, despite the fact that she had unlimited experience placed at her disposal, took too long, and the engine did not appear in time to play any appreciable part in hostilities, almost all the American aeroplanes in actual service during the war being equipped with engines of European design.

Germany, from the start, decided to restrict development almost entirely to the six-cylinder straight-line water-cooled engine, on the grounds that this type of engine, though heavy, would give the maximum of reliability and fuel economy and permit of the largest production with limited manufacturing resources. Her policy was probably right, even as events turned out, and would certainly have been right had the war proved, as she undoubtedly expected, to be of short duration.

We in England had, before the war, given so little attention to aviation that we had no experience upon which to frame a policy of any kind at the start, hence we were forced to adopt the only course possible and purchase or produce every engine we could lay our hands on, regardless of type, until we had gained the necessary knowledge and experience to enable us to proceed independently. Despite this heavy handicap it is not a little to the credit of British engineers and scientists that, by the Armistice, we had the largest production, and had ourselves evolved probably the most efficient designs, both of engines and aircraft, of any of the countries concerned.

The progress of the war very soon indicated that several entirely different types of aircraft would be needed; for example:

(1) A very fast, but small and light fighting machine, capable of rapid manœuvring and of climbing to high altitudes, but not required for long sustained flight.

(2) An observation aeroplane for spotting for artillery and generally reconnaissance work, to be capable of attaining high altitudes, and of long-sustained flight, but not necessarily very fast.

(3) A large bombing machine capable of carrying heavy loads and of flying great distances without replenishment.

The first type required an engine of high power at any altitude, light weight, and short over-all length. Economy of fuel and oil was, however, of secondary importance since such machines were not normally expected to carry out long flights.

The second type required an engine of high power at high altitudes only and high fuel economy. Since such machines always climbed to a high altitude before crossing the enemy's lines, and could therefore afford to climb slowly, the power output at or near the ground was of little importance, provided it was sufficient for safety in taking off.

The third type required an engine of the highest possible economy, both in fuel and oil, and a very high power output at or near ground level in order to enable it to take off with the heaviest possible load. Since such machines were used almost entirely by night, performance at very high altitudes was not required. If capable of leaving the ground at all at the start, they would, by the time they had reached their objective, have attained a sufficient altitude to ensure reasonable security against anti-aircraft fire from the ground.

For all these purposes Germany decided to compromise with a single type of six-cylinder straight-line engine of between 160 and 300 B.H.P., a typical example of which is shown in fig. 156, though towards the closing stages, finding that she was being outclassed by the Allies, who were employing specialized engines for each class of machine, she began to show signs of departing from this policy.

For the first class of machine the lightest and shortest possible engine was required, and this undoubtedly would have been met by the air-cooled fixed radial had any country succeeded in producing a really successful example of this type and of sufficient power. No such example was produced before the Armistice, although all the allied countries made strenuous attempts to do so. Machines of this class were therefore fitted either with air-cooled rotating engines or with water-cooled eight-cylinder Vee type.

For the second class, namely, the reconnaissance machine, the eight- and twelve-cylinder Vee type and the six-cylinder straight line were used. The latter was, however, never held in particular favour by the Allies, despite its inherent reliability.

For the heavy bombing machines, the Allies used the same engines as for reconnaissance work, for it was not until the last stages of the war that the military importance of this class of machine was appreciated, when several engines of from 500 to 800 H.P. of the twelve-cylinder Vee type were developed for the purpose.

The requirements of aerial transport in peace time call for an engine of considerable power, capable of getting off with heavy loads, but not required to climb to very high altitudes, nor to cover very great distances without replenishment of fuel. In the former respect the conditions are somewhat similar to those obtaining in

Fig. 156.—Typical Modern Six-cylinder German Aero-engine

the case of bombing machines, except that the need for economy in fuel and oil is not so insistent.

Both for bombing machines and, more particularly for commercial transport, reliability is of great importance, and, since the machines are heavily loaded and fly in a relatively dense atmosphere, the load factor on the engine is very much higher than in other types of aircraft, so that the engines are operating under much more strenuous conditions. Finally, we must consider the seaplane or flying boat, in which a very high power output is required momentarily when getting off from the water.

It is popularly supposed that an aeroplane engine operates

normally at or near its full load, but this is very far from being the case, for it must be remembered that at an altitude of about 20,000 ft. the density of the atmosphere is but little more than half that at ground level, so that, even though the throttle be wide open, the weight of charge taken into the cylinder is only about half the normal, and the conditions are very much akin to throttling down to half torque on the ground, for both the pressures and the heat flow are reduced to nearly one half. It is only when leaving the ground and climbing for the first few thousand feet that the load factor is at all high.

An average modern single-seater fighting machine will climb 10,000 ft. in about seven minutes, that is to say, within seven minutes of leaving the ground the machine is in air at an absolute pressure of only 10·6 lb. per square inch, and the indicated horse-power is only 72 per cent of that developed at ground level so that the power output is very rapidly reduced, even though the throttle be kept wide open. It is only in the heavy bombing machine and in aerial transport that the engine is called upon to work "all out" at low altitudes, and therefore under severe conditions as regards pressures and heat flow.

Discussion still rages as to the lightest form of aircraft engine consistent with sound mechanical design; and the dimensional theory has been used and abused to an unwarranted extent. The dimensional theory is applicable only when all dimensions are strictly proportional, which they can never be. According to such a theory the lightest possible engine will be the type with an infinite number of pistons connected to a minimum number of cranks. This holds good only so long as small pistons, cylinders, &c., can be made proportionately as thin as larger ones, which is out of the question in the sizes in view, and also all the auxiliary structure and mechanism can be reduced in proportion.

The nearest practical approach to this theoretically ideal form is the air-cooled radial engine with seven or nine cylinders disposed radially round a single crank, or double the number round two cranks. It does not pay to increase the number of cylinders beyond nine, because they then crowd too closely round the crankcase, necessitating a larger crankcase and longer connecting-rods for the same stroke. Though very attractive on paper, this form of engine has certain inherent defects.

(1) The loading on the single crankpin is excessive, and necessitates very special treatment. Moreover, this loading being due

almost entirely to centrifugal and reciprocating forces, does not ease off appreciably as the density of the air is reduced.

(2) The distribution of fuel and air in uniform proportions to an odd number of cylinders disposed radially is no easy problem.

(3) The valve gear is troublesome, and being so widely scattered it is practically impossible to enclose or lubricate it.

Apart from its light weight, the fixed radial engine has several important advantages which go far to balance its inherent defects.

(1) It lends itself admirably to air cooling, since every cylinder has equal advantages, and all have their combustion heads projecting well into the slip stream from the propeller.

(2) It is very short, and therefore particularly attractive from the point of view of rapid manœuvring.

(3) Its general shape and ease of attachment to the fuselage of an aeroplane are points very much in its favour.

During the war numerous attempts were made to produce such an engine, but without much success, owing to the defects named above, but since the Armistice at least two successful engines of this type have been produced, notably the Bristol Jupiter engine of 380 B.H.P. and the Armstrong Siddeley Jaguar of 350 B.H.P.

The difficulties as regards crankpin loading and distribution can both be obviated by the employment of a fixed crankshaft with the cylinders rotating round it, and by feeding the fuel and air through the crankcase as in the Gnome, Le Rhone, Bentley, and other engines. This form was widely used before and during the earlier stages of the war, particularly by France. For relatively small powers up to about 200 B.H.P. it is satisfactory, but the windage resistance becomes very serious and the gyroscopic effect due to the large rotating mass very troublesome, when the size is increased beyond this limit. A compromise between these two types wherein both the cylinders and crankshaft rotate in opposite directions has been suggested, and several engines have actually been built, notably one by Siemens and Halske in Germany, and an experimental engine of about 250 B.H.P. by Messrs. Ruston and Hornsby, built to the designs of Mr. A. E. L. Chorlton.

After the single-crank radial the next stage is the fan type, such as the well-known Napier Lion, in which three pistons operate on each crank, and the Maltese cross type with four cylinders per crank. It is usual to make both these types with blocks of four cylinders, making twelve or sixteen in all; though some few examples have been built with six-throw cranks, making eighteen and twenty-

four cylinders respectively. Figs. 157 and 158 show the Napier Lion engine, an example of this type which has proved particularly successful.

The next type is that in which two pistons are coupled to each crank, generally known as the Vee-type engine. This type has usually either four or six cranks, and therefore eight or twelve cylinders. To the Vee-type class belong most of the successful engines used by the Allies during the war, and probably also at the present day, though more recent developments have brought both the single-crank radial and the straight-line into prominence. The

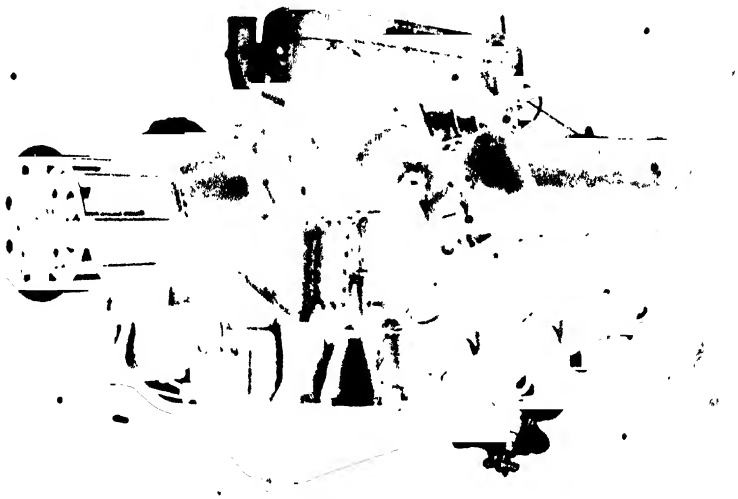

Fig. 157.—Napier Lion Engine

Rolls-Royce Eagle and Falcon engines, the Hispano Suiza, and the R.A.F. were among the most successful examples of this type used during the war, while the Rolls-Royce Condor of 550 B.H.P., the Liberty of 400 B.H.P., and the 600 H.P. Fiat represent excellent examples of more modern development.

Finally, we have the plain straight-line six-cylinder engine with one piston operating each crank ; a type which has been vehemently condemned by the supporters of the dimensional theory, but which was used practically throughout by the Germans, and to a considerable extent by the Allies also, e.g. the Siddeley Puma and the Beardmore, both of which engines did admirable work and competed very favourably with the other types.

Controversy as to the lightest type of engine will probably continue to rage indefinitely, since so many conflicting and often indeterminate factors have to be taken into account, viz. reliability, fuel and oil consumption, &c. On the grounds of reliability there can be no doubt but that for equal excellence of design and workmanship the advantage lies with the straight-line six-cylinder, since the load factor on its bearings is considerably the lightest, the stresses are for the most part simple and direct, and can

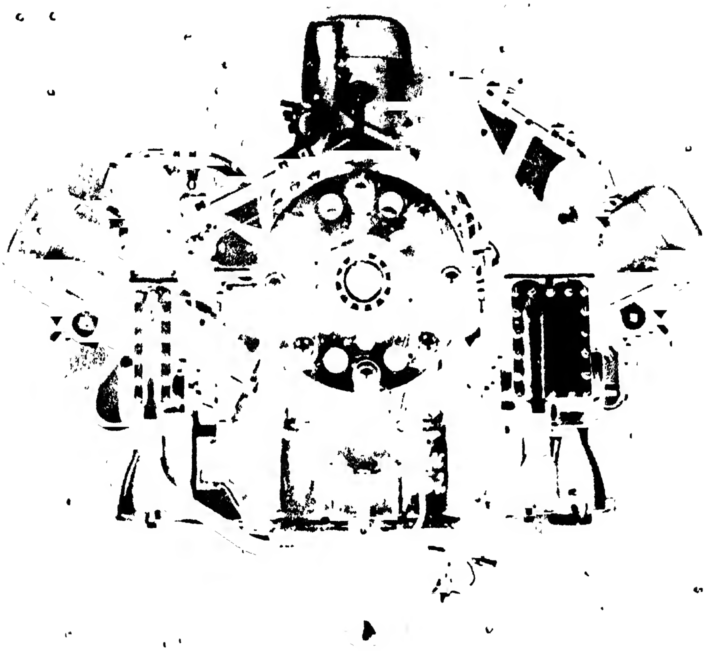

Fig. 158.—Napier Lion Engine, End View

be dealt with by simple and direct means; also the auxiliary gear, upon which the reliability of the engine so largely depends, is reduced to the minimum. On the score of fuel efficiency it has again, for equal excellence of design and workmanship, all the advantage, since the individual cylinders are larger and the losses therefore less in proportion, while, having only two carburettors and an inherently good form of distribution, the losses due to defective carburation and distribution can, with a given amount of superintendence, be kept lower than with any other type. Finally, since the load factor

on its bearings is the lowest, less oil need be circulated for cooling purposes, and consequently less is thrown on the cylinder walls and consumed. On all these counts, therefore, the straight-line six has advantages which go far to compensate for the extra material in the crankshaft and crank chamber as compared with other types. It is, of course, impossible to evaluate the factor of reliability and to equate it in terms of weight, though, clearly, reliability is always worth some pounds in initial weight.

Again, in comparing the weights of various engines the efficiency of the propeller is sometimes overlooked. One of the essential requirements of any aero-engine is that it shall turn the propeller at its most efficient speed, and this, in the case of large and heavily loaded machines, is a comparatively low one: Hence the engine must either turn at a relatively low speed, 1200-1400 R.P.M., or must be geared down, if a high over-all efficiency is required. Experience has shown that the weight of reduction gearing very nearly balances the increased weight of engine required to develop the same power at the lower speed, while here again the factor of reliability looms largely, since reduction gears, at the best of times, are a source of weakness, the more so as the ratio of reduction is increased. If we require the highest over-all efficiency, from the fuel burnt to the thrust of the propeller, and assume a reasonable duration of flight, we find that the over-all weight of the power unit, together with its fuel or oil, becomes, in practice, virtually a function of the piston speed, and this almost irrespective of number or disposition of cylinders or of the use or otherwise of reduction gearing.

Air- or Water-cooling.—Here again a great deal of controversy rages as to which is the more desirable. The water-cooled engine starts with the heavy handicap of a radiator and water connections, involving considerable additional weight, and, what perhaps is even more serious for military purposes, much greater vulnerability; but against these defects must be offset a very large advantage on the score of reliability, and the ability, owing to the lower cylinder temperature, both to consume less oil, to employ a higher compression, and therefore to obtain a lower fuel consumption. It is not proposed to deal, at any length, with the pros and cons of air- *versus* water-cooling, but it is probably sufficient to point out that the radial engine, by reason of the disposition of the cylinders and its relation to the slip stream from the propeller, offers the most ideal case for air-cooling, and, the author is tempted to think, the only case for it.

Controversy as to the lightest type of engine will probably continue to rage indefinitely, since so many conflicting and often indeterminate factors have to be taken into account, viz. reliability, fuel and oil consumption, &c. On the grounds of reliability there can be no doubt but that for equal excellence of design and workmanship the advantage lies with the straight-line six-cylinder, since the load factor on its bearings is considerably the lightest, the stresses are for the most part simple and direct, and can

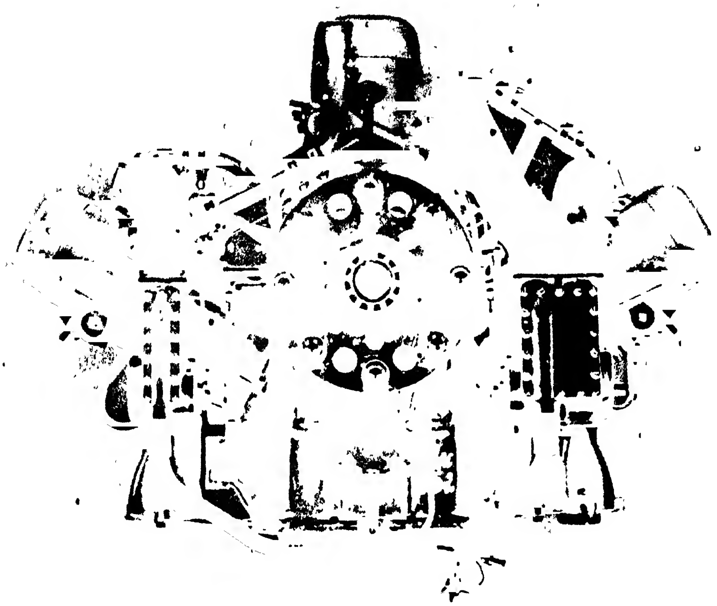

Fig. 158.—Napier Lion Engine, End View

be dealt with by simple and direct means; also the auxiliary gear, upon which the reliability of the engine so largely depends, is reduced to the minimum. On the score of fuel efficiency it has again, for equal excellence of design and workmanship, all the advantage, since the individual cylinders are larger and the losses therefore less in proportion, while, having only two carburettors and an inherently good form of distribution, the losses due to defective carburation and distribution can, with a given amount of superintendence, be kept lower than with any other type. Finally, since the load factor

that under normal working conditions local over-heating can be avoided and results obtained comparable with those of a water-cooled cylinder.

It has, however, been found very difficult to ensure a sound job when casting the head on to a steel barrel, while the alternatives of bolting, screwing, or shrinking have none of them proved sufficiently reliable. Probably casting on to the barrel is the most hopeful, for it is not impossible, and the difficulties which have been encountered are mostly questions of foundry technique, and, as such, will probably be surmounted as more experience is gained. Given that, as in a radial, or rotary engine, the position of the cylinder is such that it has the best of facilities for cooling, it has been demonstrated that the air-cooled cylinder will give from 90 to 95 per cent as high power and efficiency as a corresponding water-cooled cylinder, so long as there is no distortion, leakage, or detonation; but while the water-cooled cylinder can generally survive such ailments due to the automatic intensification of heat transference from the seat of the trouble, the air-cooled cylinder has no such advantage, and must quickly give way to local overheating followed by pre-ignition, and perhaps also by distortion and seizure of the piston or burning out of the valves.

The lack of recuperative power which in the author's opinion is the weakness of the air-cooled engine applies also, though to a lesser extent, to those water-cooled engines in which there is a double metal wall through which the heat has to pass before reaching the cooling water. In fig. 165 is shown a part-sectioned cylinder block of the Hispano Suiza water-cooled engine, in which it will be seen that a composite construction is used consisting of a complete aluminium cylinder into which is screwed a steel thimble forming both the liner and cylinder head, and that the valves seat directly on to this steel thimble. This form has many important advantages from a constructional point of view, but it is open to the objection that the heat has to pass through two separate thicknesses of metal, in contact only by screwing, before reaching the cooling water. Given good fitting this suffices for the normal rate of heat flow, but it has a very much reduced margin of safety for dealing with excessive rates of flow such as occur when detonation is set up, &c.

Cylinder Construction.—In an aircraft engine it is necessary always to provide some form of composite cylinder construction, because the limitations of weight deny the use of the normal construction in which the outer jacket is cast in one with the cylinder

liner and of the same material. Resort must, therefore, be had to composite built-up forms, and much diversity of opinion prevails as to the relative merits and demerits of the different forms in use.

Fig. 159 shows the cylinder construction used in the 260 B.H.P. German Mercedes engines such as were fitted to the large Gotha bombing planes used during the war for long-distance bombing raids.

Fig. 159.—Cylinder Construction used in the 260 H.P. Mercedes Engine

Fig. 160.—Austro-Daimler Cylinder

In this case a high-carbon steel barrel is screwed into a pressed or cast-steel head, the bottom edge of which is spun over one of the flanges of the barrel and welded to it in order to ensure against any possible leakage down the thread. A light built-up sheet-steel jacket is then welded over all. This form of construction has proved a very reliable one, provided always that the welding has been skilfully accomplished. It is open to the objection that it requires a good deal of specialized plant and special skill in welding, especially

in dealing with the sparking plug losses and the water connections. Given, however, the requisite skill and plant, it is certainly a very satisfactory method. The long and very well-cooled guides for the exhaust valves are an excellent feature and deserve special notice.

Fig. 160 shows the form of construction used in the early Austro-Daimler, and later in the 120 and 160 B.H.P. Beardmore six-cylinder engines. In this design the cylinder barrel and head are cast in one piece in cast iron, but the inlet valve is fitted in a separate detachable housing held in place by an annular locking nut. The lower end of the cylinder barrel is screwed externally to receive a steel flange for the holding-down bolts.

The water-jacket is of electro-deposited copper; this is formed in place on a wax matrix, which is subsequently melted out. Much experience and great precautions are needed to ensure a uniform deposition of copper and thorough adhesion to the cast iron. In this case also, once the plant is available and the necessary experience has been gained the method is very satisfactory.

Fig. 161 shows the form adopted by the Maybach Company in Germany for their 300 B.H.P. six-cylinder engines used in the later Zeppelin airships and, during the latter phases of the war, in many of the larger aeroplanes.

In one form of this construction the cylinder head, together with the whole of the water-jacket, is of cast iron, and a high-carbon steel barrel is screwed and sweated, but not welded, into the head, the lower end of the jacket being sealed by means of a rubber ring. In another form, only the cylinder head and the upper portion of the jacket are of cast iron, the jacket of the barrel being a very light seamless steel tube also screwed to the cylinder head in the same manner as the liner.

The construction shown in fig. 162 is that used by the Benz Company for all their aero-engines. In this case the whole of the cylinder barrel, cylinder head, and holding-down flange are cast in one piece in cast iron over which a light pressed steel jacket is electrically welded direct on to the cast iron. The welding of such a thin steel jacket to a relatively thick cast-iron body is no easy problem, but it has been met satisfactorily by this company. Attention should be called to the use in these engines of a special support from the crown of the piston to the gudgeon-pin in order to transmit the load as directly as possible to the connecting-rod.

The construction shown in fig. 163 is that adopted by Messrs. Rolls-Royce, and subsequently employed in the Liberty and several

other engines. In this case the cylinder barrel and head are forged in one piece, and the inlet and exhaust valve elbows are screwed and welded into place, the whole body being subsequently covered by a light pressed steel jacket, welded over all.

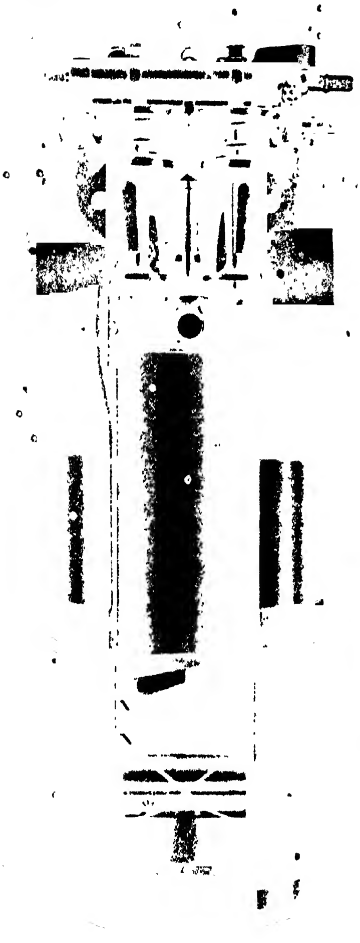

Fig. 161.—Zeppelin Airship Engine Cylinder

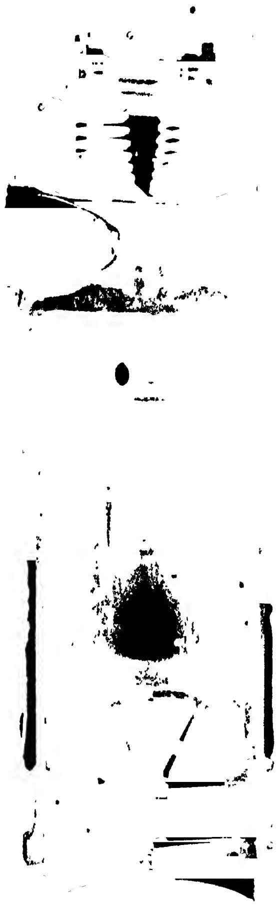

Fig. 162.—Benz Cylinder

Fig. 164 shows the cylinder construction used in the earlier Sunbeam engines, in which complete blocks of cylinders were cast in iron together with their cylinder heads and the upper portion of the water-jackets; to save weight the whole of the sides of the casting below the valve outlets are removed and replaced by light sheet

metal plates. In some of the later Sunbeam engines such as the 200 B.H.P. Arab engine, the complete cylinder block was cast in aluminium with thin steel liners shrunk in.

Fig. 165 shows the construction of the Hispano Suiza cylinder block already referred to, in which a complete steel thimble, forming

Fig. 163 —Rolls-Royce Cylinder

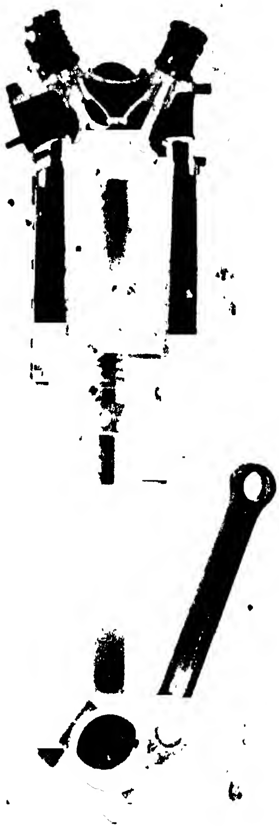

Fig. 164.—Sunbeam Cylinder

both the cylinder liner and valve seats, is screwed directly into an aluminium cylinder block.

In the Siddeley Puma engine shown later in fig. 175, blocks of three-cylinder heads, together with the upper portion of the water-jacket, are cast in aluminium with pressed-in bronze valve seats. Into these are screwed, for a short length only, thin steel cylinder barrels. The lower portions of these exposed steel liners are

enclosed by means of light and very thin die-cast aluminium jackets which are bolted direct to the cylinder head casting, while the lower joints consist of rubber-packed stuffing glands.

The form of construction shown in the previous chapter, in connection with the three-litre Vauxhall racing-car engine, has been used in several experimental aero-engines, and has been adopted

Fig. 165.—Hispano Suiza Cylinder

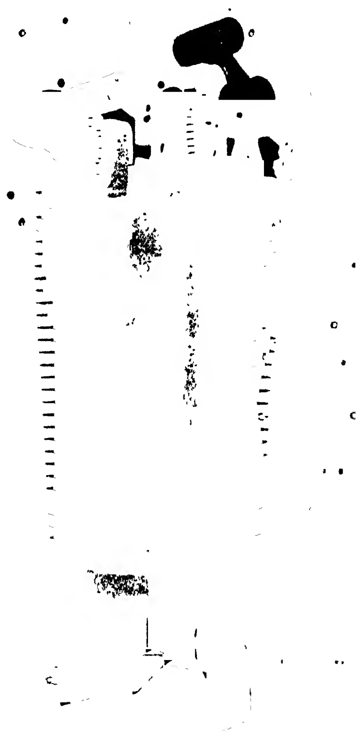

Fig. 166.—Clerget Cylinder

by Messrs. Beardmore in the large 750 B.H.P. six-cylinder aero-engine built by that firm.

It has in the author's opinion much to recommend it, not the least being its extreme simplicity and ease of manufacture.

Turning now to air-cooled engines the problem becomes somewhat different, since weight is no longer the sole consideration, high conductivity being at least equally; if not more important. In engines with rotating cylinders the cooling conditions are very favourable, and in such engines it is usual to employ plain steel

cylinders machined throughout from a single forging as shown in fig. 166, which is the form employed in the Clerget rotating engine, and fig. 167, which shows a section of the Gnome single valve engine.

Fig. 168 shows a section of the Le Rhone cylinder. In this case the whole of the cylinder and head is machined from a single piece of steel, but a very thin cast iron liner about 1 mm. in thickness is

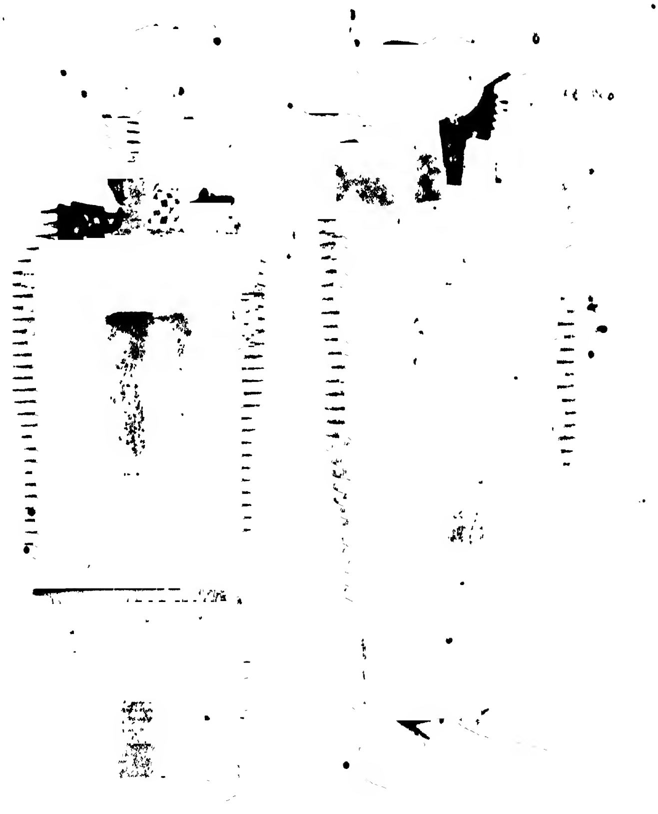

Fig. 167.—Gnome Cylinder Fig. 168.—Le Rhone Cylinder

pressed in. This construction is curious, and the author has never been able to discover for what reason it has been adopted.

In the B.R. 1 and B.R. 2 rotating cylinder engines, fig. 169, the cylinder barrel consists of a hard steel liner surrounded by a thin and light ribbed aluminium jacket. The cylinder heads in this engine are detachable and are of steel, also a curious construction.

In the case of fixed cylinder engines the problem of cooling

becomes more difficult and resort had to be made to more complicated constructions.

Fig. 170 shows an experimental R.A.E. cylinder for fixed cylinder air-cooled engines. This consists of a thick aluminium casting with deep ribs and with steel valve seats cast in position. A thin steel liner is shrunk in as shown in the photo of the sectioned cylinder.

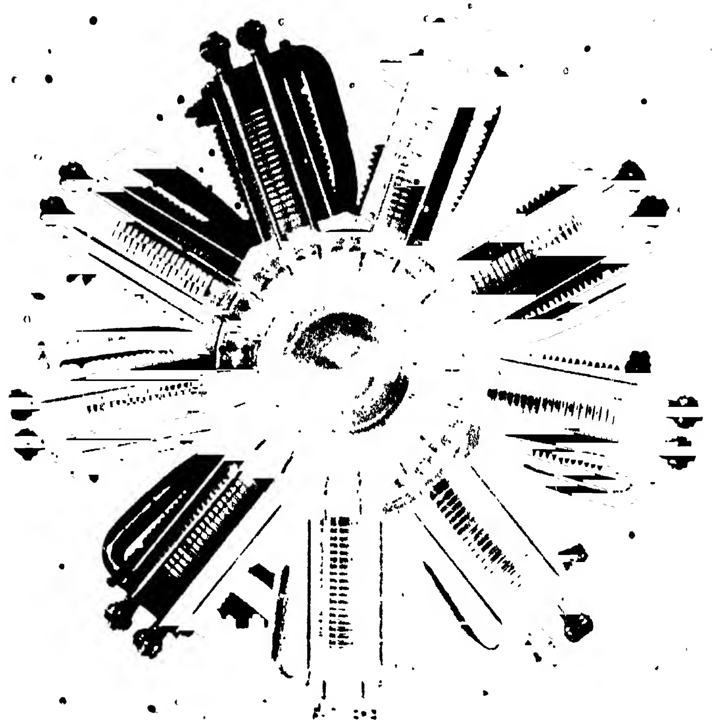

Fig. 169.—9 Cylinder B.R. Rotating Engine

This proved satisfactory for a time, but contact between the liner and cylinder body gradually failed, resulting in overheating of the liner. As explained previously, this form of cylinder construction was subsequently abandoned in favour of one consisting of an aluminium cylinder head cast on to a plain ribbed steel barrel.

In the Bristol Jupiter engine shown in fig. 171 the whole of the cylinder barrel and head are of steel, but a cast aluminium poultice containing the inlet and exhaust valve elbows and heavily

ribbed, is attached to the flat steel head. This form is simple to manufacture, and has the advantage that if contact between the cylinder-head and aluminium poultice is impaired by warping, it can be restored by scraping the surfaces.

As stated previously, no satisfactory fixed radial air-cooled engine of adequate size was developed during the period of the war despite the most strenuous efforts in this direction. Since the war,

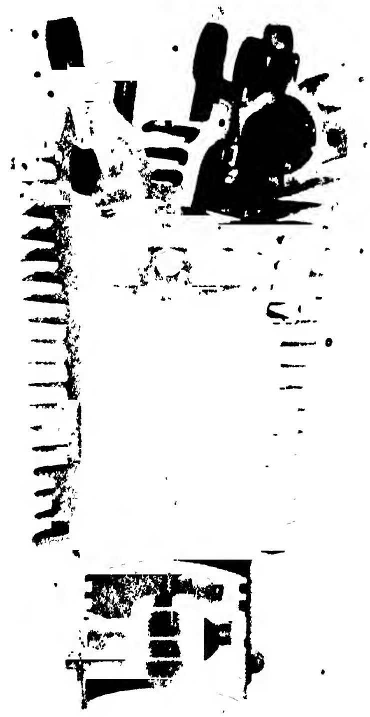

Fig 170.—Experimental R.A.E. Air-cooled Cylinder

however, two such engines have been developed, namely, the Bristol Jupiter and the Siddeley Jaguar, figs. 170-174.

The former is a single crank nine-cylinder engine developing a normal power output of 380 B.H.P. at 1575 R.P.M. The cylinders are of 5·75-inch bore with a stroke of 7·5 inch. The normal compression ratio of this engine is 5 : 1, and it will run continuously for long periods at a brake mean pressure of 109 lb. per square inch with a consumption of 0·535 lb. of petrol and 0·048 lb. of oil per B.H.P.

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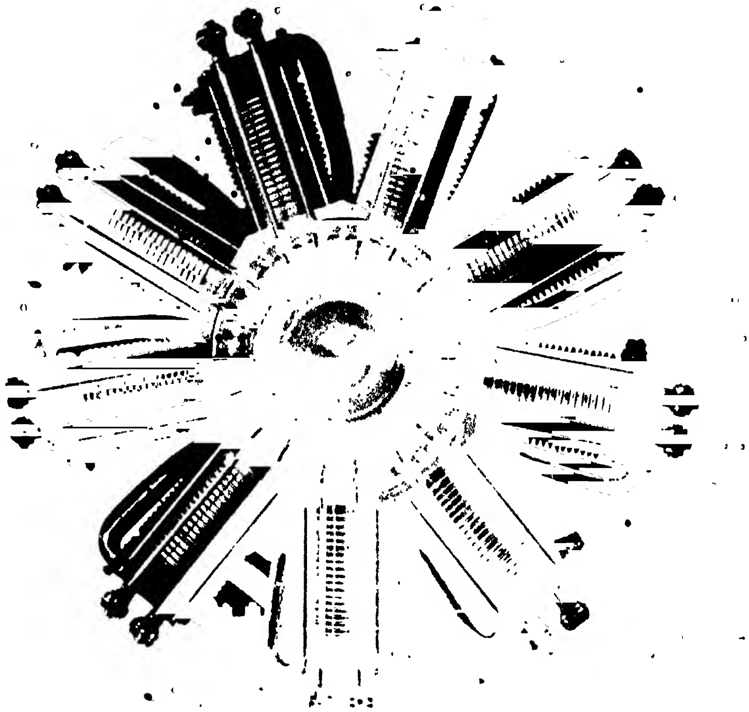

Fig. 169.—9 Cylinder B.R. Rotating Engine

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In the Bristol Jupiter engine shown in fig. 171 the whole of the cylinder barrel and head are of steel, but a cast aluminium poultice containing the inlet and exhaust valve elbows and heavily

are taken to deal with the very heavy loading, due to the combined centrifugal and inertia pressures from the nine pistons and rods all operating on a single pin.

One of the greatest difficulties encountered with air-cooled engines is that of dealing with the expansion of the cylinder and the resulting increase in the valve motion clearances when hot. In the case of the Bristol Jupiter engine this has been dealt with most

Fig 172.—400 H.P. Bristol Jupiter Radial Engine

effectively by means of a very ingenious compensating device developed by Mr. Raymond Morgan. Briefly, this consists in the provision of a movable fulcrum pin for the overhead valve rockers which is controlled by a fixed rod attached at one end to the crank-case, and at the other to a hinged cradle carrying the valve rockers. This control rod being subject to the same temperature conditions as the operating rods maintains the same relative length, with the result that as the hot cylinder expands it tends to draw the cradle

down, and so to retain the same tappet clearance at all cylinder temperatures. These control rods can be seen on the forward end of the engine (see fig. 172).

The Siddeley Jaguar shown in figs. 173 and 174 is a fourteen-

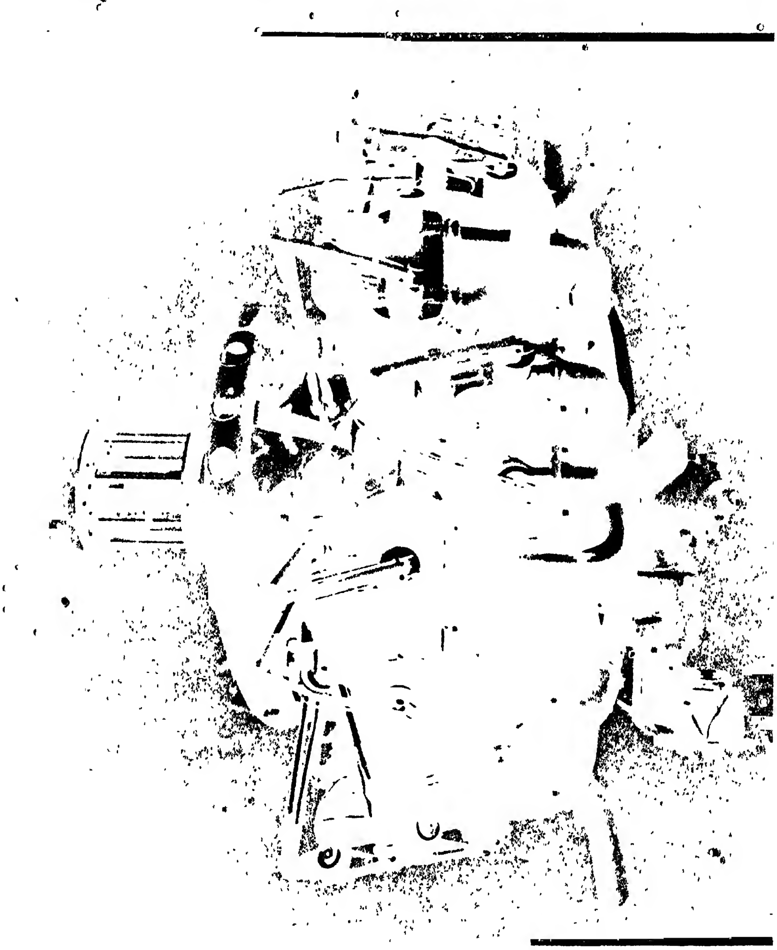


Fig. 173.—Siddeley Jaguar Engine

cylinder two-crank radial, and develops a normal power output of 350 B.H.P. at a speed of 1500 R.P.M.; the cylinders are five-inch bore and five-and-a-half-inch stroke.

As in the case of the Bristol Jupiter this engine also has been developed since the Armistice, though in both cases the earlier

stages of development were in progress during the war. The manufacturers give the fuel and oil consumption of this engine as 0·525 and 0·027 lb. per B.H.P. hour respectively, a gross consumption of 0·552 lb. per B.H.P. hour. The weight of the engine alone is given as 710 lb. or 2·03 lb. per horse-power. The gross weight with fuel and oil for six hours' flight (exclusive of tanks), works out at 1870 lb. or 5·35 lb. per horse-power. In this engine, the cylinder barrels are of steel as in the Jupiter, but the cylinder heads are aluminium

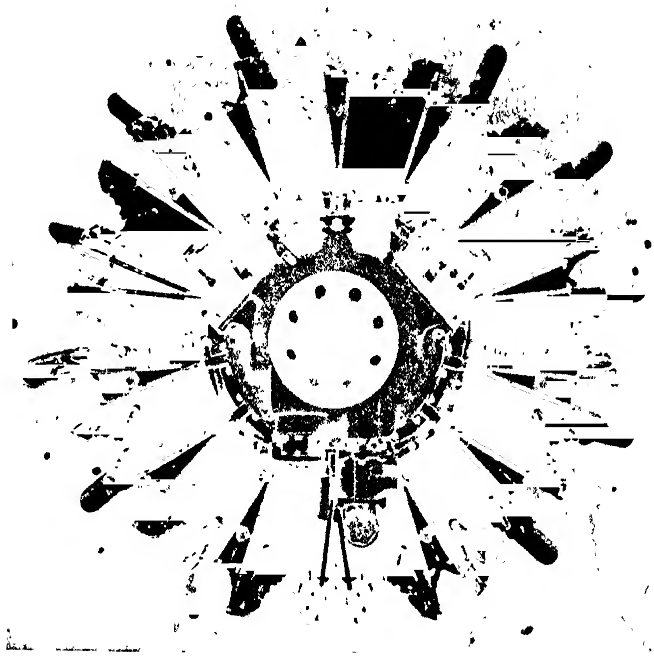


Fig. 174.—Siddeley Jaguar Engine

castings screwed on to the steel barrels. Ignition is by high tension coil and battery, and a small dynamo is provided for charging the accumulator.

The engine shown in figs. 175 and 176 was known during the war as the Siddeley Puma, and is a development of the B.H.P. engine designed by Messrs. Beardmore and Major Halford of the R.A.F. It has six cylinders each of 145 mm. bore and 190 mm. stroke with a normal power output of 240 B.H.P. at a speed of 1400 R.P.M. With a compression ratio of 5 : 1 the fuel consumption is 0·5 lb. and

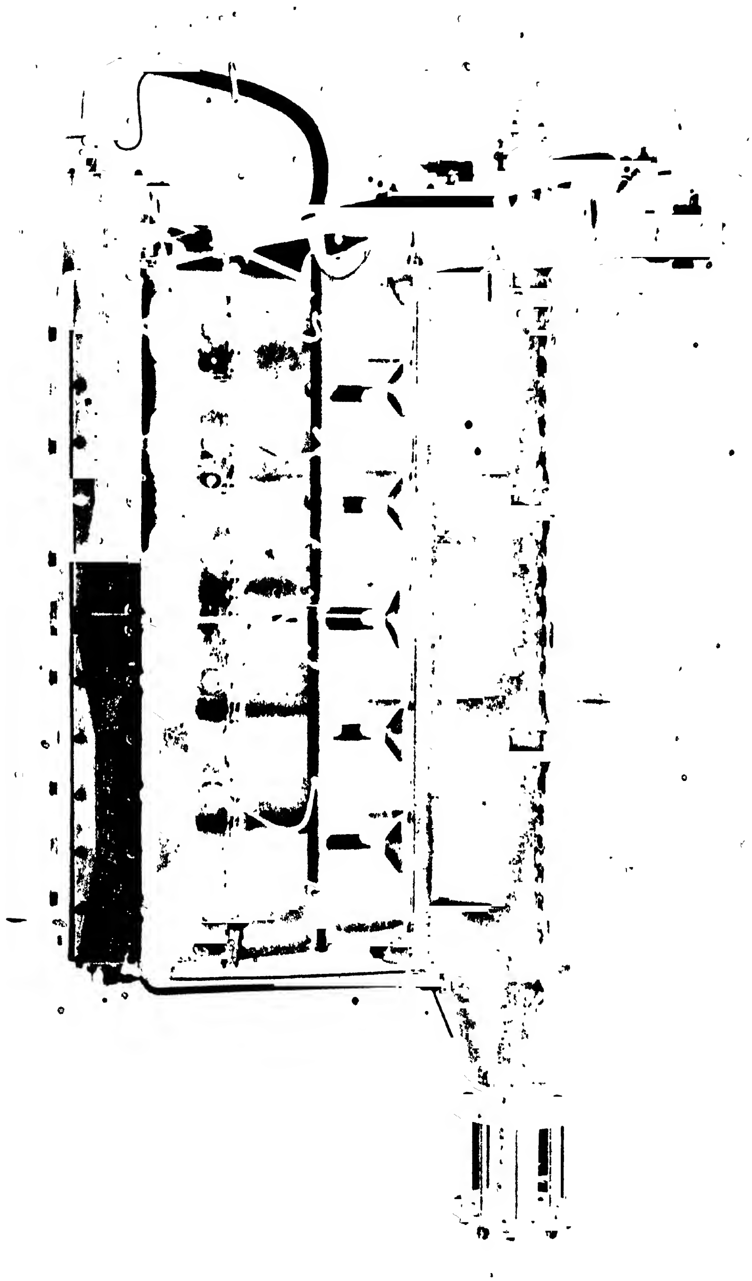


FIG 175.—Siddeley Puma Engine

the oil consumption 0·03 lb. per B.H.P. hour, a gross consumption of 0·53 lb. per B.H.P. hour. The weight of the engine complete with radiator and cooling system is 731 lb. or 3·25 lb. per horse power. The gross weight with fuel and oil for six hours (exclusive of

Fig. 176. Siddeley Puma Engine

tanks) amounts to 1443 lb. or 6·01 lb. per horse-power. This engine is essentially a plain and straightforward piece of design, simple alike in manufacture and in handling.

The engine shown in figs. 157 and 158, also in the sectional drawings figs. 177 and 178, is the Napier Lion. It has twelve cylinders

each of five-and-a-half-inch bore and five-and-one-eighth-inch stroke arranged in three groups of four cylinders operating on

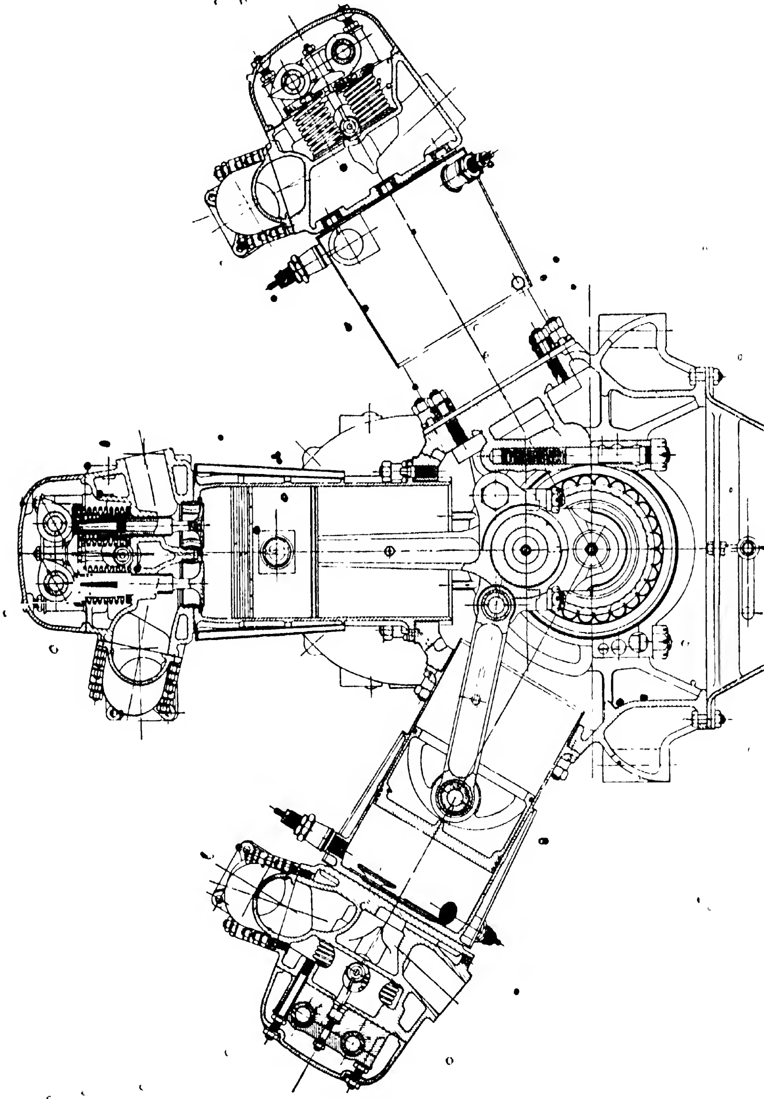
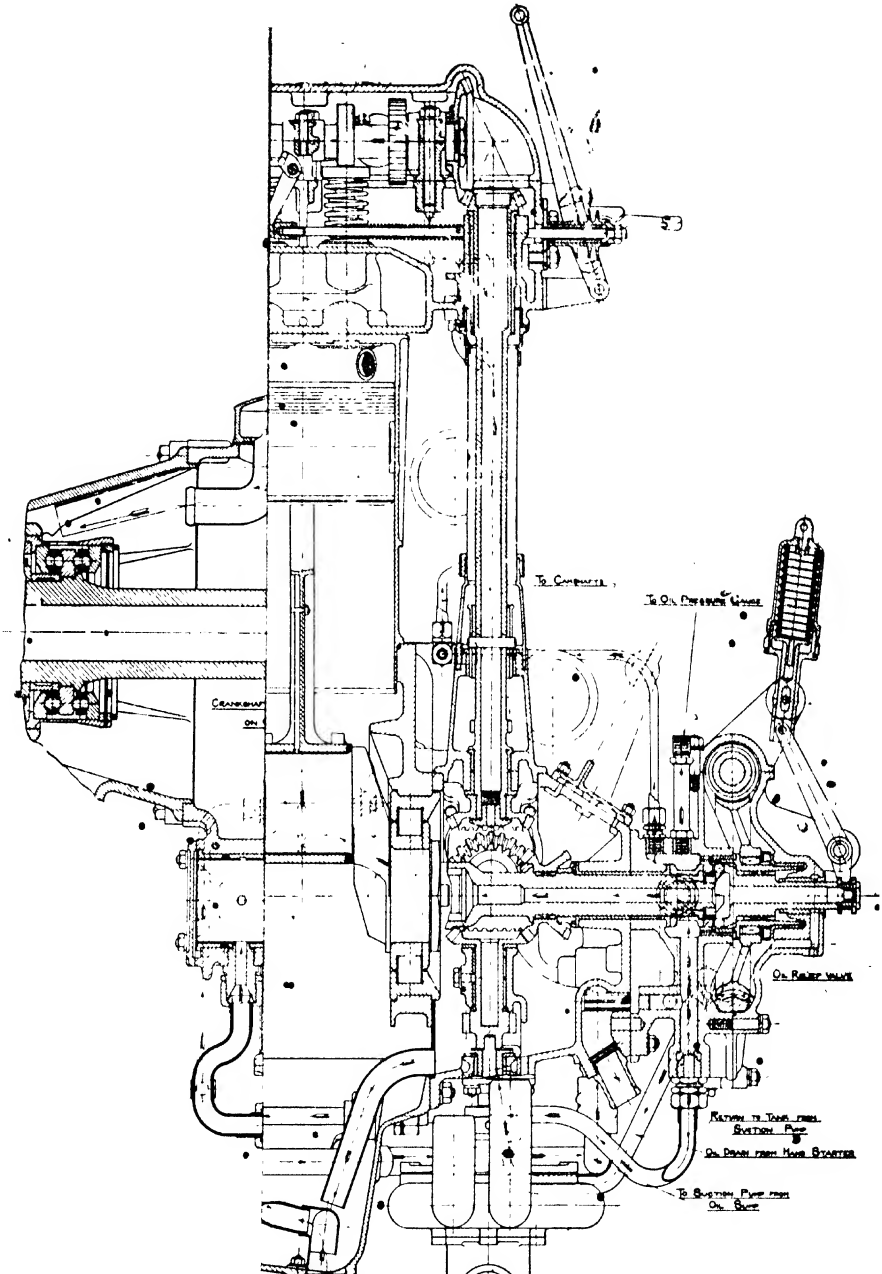


Fig. 178.—Section of Napier Lion Engine

a plain four-throw crank. It develops 450 B.H.P. at its normal crankshaft speed of 2000 R.P.M. Its weight complete with radiator, cooling system and speed reduction gearing is only 1134 lb. or



To Camshafts
To Oil Pressure Gauge
Oil Relief Valve
Return to Tank from Suction Pump
Oil Drain from Hand Starter
To Suction Pump from Oil Sump

2·52 lb. per horse-power. Its fuel and oil consumption per B.H.P. hour are given as 0·495 and 0·022 lb. respectively.

The gross weight of the engine complete with all necessary gear and fuel and oil for six hours' flight is 2534 lb. or 5·63 lb. per horse-power.

In fig. 179 is shown the 350 B.H.P. Rolls-Royce Eagle engine, a twelve-cylinder Vee type, having cylinders of four-and-a-half-inch bore and six-and-a-half stroke, with a normal crankshaft speed of 1800 R.P.M. This engine, which was developed by Messrs. Rolls-Royce during the war, proved to be undoubtedly the most satis-

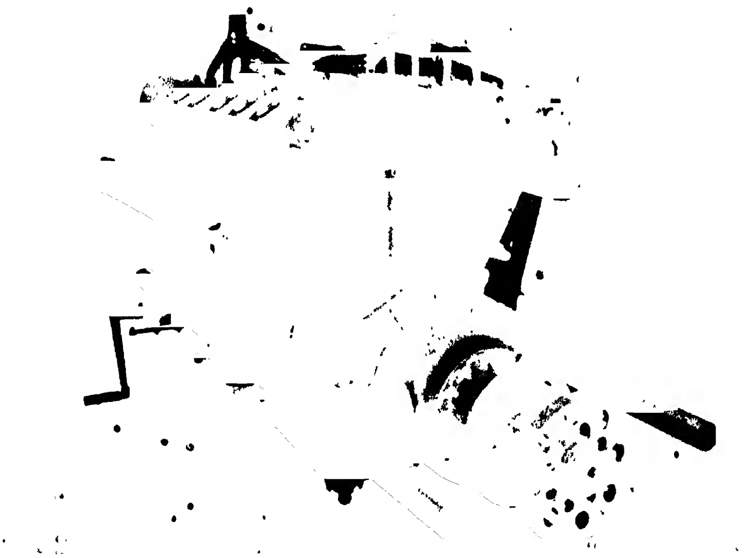


Fig. 179.—Rolls-Royce Eagle

factory and reliable engine in the hands of the Allies, and was of great value, not only on account of its magnificent performance, but perhaps even more because of its encouraging effect on the *moral* of the Allied pilots. Official records compiled in France during the war show that the average number of hours flown by these engines between overhauls was 103·2, or very nearly double that of any other aero-engine used in the British service. This engine, also, is of interest because it is at once probably the most complicated and quite the most reliable engine yet built for aircraft.

Its weight complete with epicyclic speed reduction gear, radiator

Fig. 180.—600 H.P. Fiat Engine

Fig. 181.—1000 H.P. Napier Cub Engine

cooling system, &c., is 1177 lb. or 3·37 lb. per horse-power. The fuel and oil consumption are given as 0·50 and 0·028 lb. per B.H.P. hour respectively, so that the weight complete for a six-hour flight is 2287 lb. or approximately 6·5 lb. per horse-power.

In fig. 180 is shown the 600 B.H.P. Fiat twelve-cylinder Vee engine, which also may be taken as a fairly typical example of the

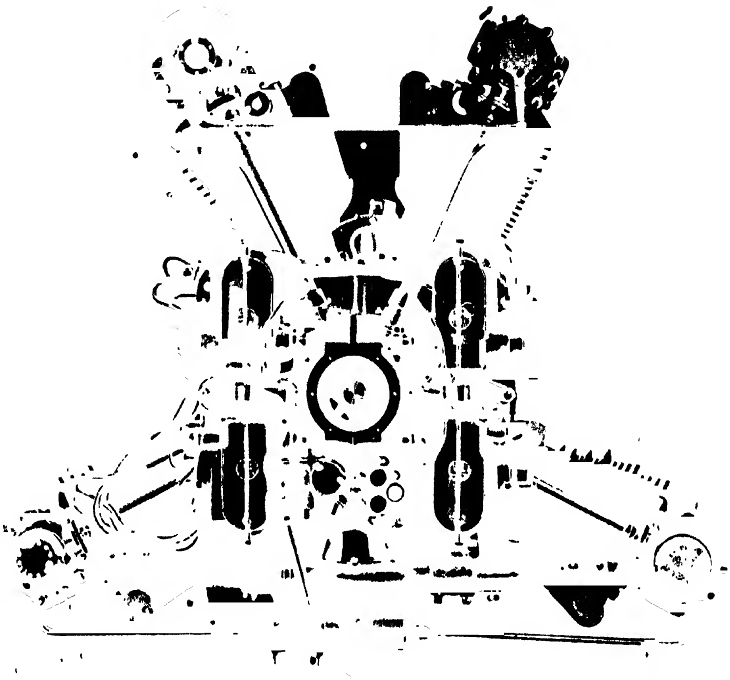


Fig 182 —1000 H.P. Napier Cub Engine

type of engine developed by the Allies during the latter phases of the war. In figs. 181 and 182 are shown photos of the 1000 H.P. Napier Cub engine, probably the largest engine which has yet flown successfully.

The engine shown in fig. 183 is the 500 B.H.P. Benz engine; it is of particular interest because it marks the departure from the apparently settled German policy to adhere to the straight line six-

cylinder engine for military purposes. During the closing phase of the war, the Germans evidently began to find that their policy could not be adhered to in face of the very large engines which were

Fig. 1[illegible]3.—300 H.P. Benz Engine.

being developed by the Allies, and this and a few other similar engines in course of development during 1918 bear evidence that they contemplated paying the Allies the compliment of following their lead.

Aero-engines for High Altitudes.—As a broad general-

ization the means of retaining the power output of aero-engines at high altitudes may be divided into two groups, one in which the power output is maintained by maintaining, or nearly maintaining, ground level density of the charge in the induction system, and the other, in which the density in the induction system is not so maintained, but the relative power output at the lower densities is increased by increasing the expansion ratio and so getting more useful work from a given weight of charge. In other words, the former system aims at operating with an artificially dense atmosphere at high altitudes, and the other caters for an artificially attenuated atmosphere near the ground, advantage being taken of the lower density to employ a greater expansion ratio and to obtain thereby an increase in thermal efficiency. The former system affords a means of maintaining the power output at any altitude at present attainable, though at some cost in efficiency. The latter provides only for a limited maintenance of power; but, on the other hand, it affords a considerable gain in fuel economy.

In addition to these two general systems there are also certain possible compromises between the two, which will be considered later.

With regard to the first method, that of increasing artificially the density in the induction system, this can best be accomplished by the use of a turbo-blower driven either mechanically from the main engine, by a separate engine, or by means of an exhaust driven turbine. Such a system has the advantage that the full ground level torque can be maintained at almost any height, for the limit is set solely by the mechanical strength of the engine and by its capacity for getting rid of the heat generated in a highly attenuated atmosphere. It necessitates, however, the use of a variable pitch propeller. It is possible to obtain, at high altitudes, an actual power output in excess of that developed at ground level, for it is obvious that if the weight of air per cycle be maintained, the torque also will be maintained, and since the external resistance to the rotation of the propeller diminishes, the engine will run at a higher speed and therefore develop a higher power output, even after deducting the power required to drive the turbo-blower. Under such conditions, however, the flow of heat to the cylinder jackets, &c., is increased, while the capacity of the radiator or cooling fins is reduced, at all events while climbing, owing to the reduced density of the surrounding air, and, though the lower air temperature tends to balance this to a limited extent, it is necessary to provide a much

larger radiator. At first glance this system of direct supercharging appears to afford the simplest and easiest solution of the problem, but on closer examination it will be found to present many difficulties.

In the first place, the efficiency of the best turbo-blowers, though relatively high, is actually only about 55 per cent to 60 per cent, so that the power absorbed by them is a very considerable item, especially when considered in terms of fuel consumption. In the second place, such blowers, whether driven mechanically or otherwise, must necessarily run at a very high speed, generally from 20,000 to 30,000 R.P.M., and this, in itself, introduces very serious mechanical difficulties.

Thirdly, it is of course absolutely necessary to balance the pressure in the carburetter, float chamber, &c., and to deliver the fuel against the increased pressure; also it is important to guard against leakages anywhere in the induction system or around the valve stems. This condition of affairs is of course not impossible of achievement, but it involves a good deal of added complication in the first instance, and is very difficult to maintain in service.

Fourthly, in addition to the added weight of the blower, its driving mechanism and attendant pipe-work, there is also the increased weight of radiator to be taken into account; again it is necessary to provide an additional radiator to cool the air after compression in the turbo-blower; and last, but not least, the increased stresses in the engine, both heat stresses and mechanical, must reduce greatly the reliability of the engine itself; for a normal aero-engine is not expected to develop its ground-level horse-power continuously, and although it may do so under favourable conditions on the test bed, its margin of safety is usually very small, and its reliability generally varies about inversely as the square of the power output.

While the earlier experiments with supercharging were for the most part carried out with mechanically-driven blowers, opinion in this country appears to be veering in favour of exhaust-driven blowers such as those shown in figs. 184, 185, and 186, partly because of the many difficulties encountered in the actual mechanical drive, and partly because the exhaust-driven turbo-compressor can the more readily be adapted to meet the varying conditions as regards atmospheric density, since its speed is not directly dependent upon that of the main engine. So far as purely

mechanical problems are concerned, it is doubtful whether it is easier to operate an exhaust-driven turbine at, say, 30,000 R.P.M. in an atmosphere of exhaust products at a temperature

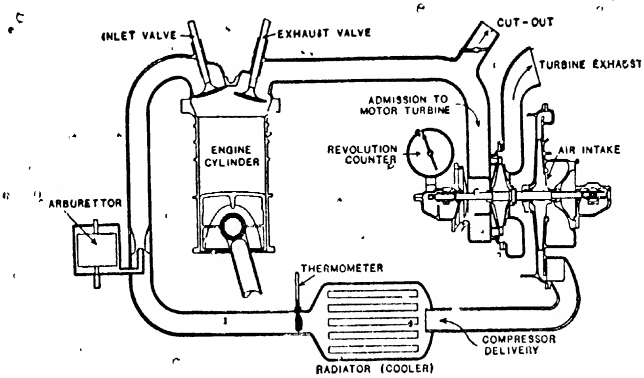


Fig. 184.—Arrangement of Turbo-compressor Supercharger

about 1100°-1200° F. than it is to drive a clean and cool air blower by suitable gearing at the same speed. The over-all efficiency of a combined exhaust turbine and blower cannot very well be determined, but at best it must be low, for, on purely mechanical grounds, it cannot operate at a very high temperature or at a sufficiently high speed, with the result that a serious proportion of the work done in compressing the air must appear as back pressure on the pistons of the main engine. Further, the resistance to the free flow of the exhaust products results, even at the best efficiencies so far attained, in the retention in the combustion chamber of hot residual products at a pressure considerably in excess of that of the entering charge (actually the best results so

Fig. 185.—Turbo-compressor for Supercharging Aircraft Engine

far obtained show an increased back pressure of about 3 lbs. per square inch above the air-pressure). Also the very serious problem of cooling the air after compression is still further aggravated by the addition, by conduction, of some heat from the exhaust turbine. At the best of times the removal of a large amount of relatively low temperature heat is a troublesome problem, involving large radiating surfaces and therefore increased weight and head resistance. The exhaust-driven blower has, however, one outstanding advantage over the mechanically driven, namely, that the variation in the impeller speed at different densities is effected automatically. With regard to mechanical driving, the chief difficulties which arise are those due to cyclical changes in the angular

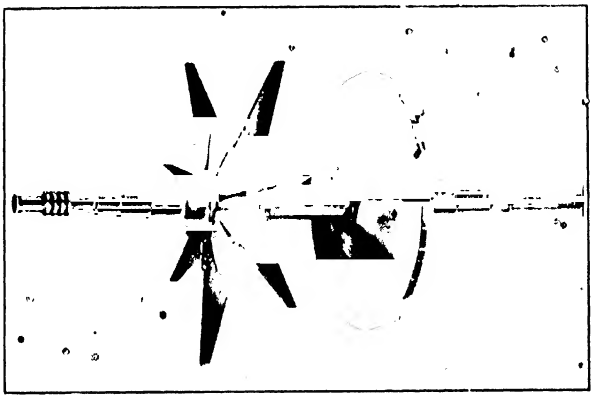


Fig. 186.—Rotor for Turbo-compressor for Supercharging Engine

velocity of the tail end of the crankshaft, from which the blower is usually driven; to sudden changes in the mean speed of the crankshaft, due to throttling down or opening up suddenly; and to faulty alignment, due to the blower not being built as an integral part of the crankcase. Most of these difficulties can probably be overcome by the provision of suitable dampers, flexible couplings, &c.

When, as in some of the large German machines, the blower is driven by means of a separate engine devoted to that one task alone, most of the mechanical difficulties disappear, and although at first sight it may seem very cumbersome, costly, and heavy to use a separate engine, yet there is a good deal to be said in favour of it, at all events in the case of large installations.

Direct supercharging is possibly the only possible means of obtaining any really large increase in power at high altitudes, and, as such, it is extraordinarily valuable; but, however it be applied, it is neither simple nor easy. For very high altitudes, it is probably quite as important to apply supercharging to the aviator as to the engine; and when supercharging is employed, the possibility of enclosing both the pilot and the engine in a light pressure tight casing is worth considering seriously, for both are equally in need of oxygen.

There is another system of supercharging, of which the author is in favour, in that it involves very little additional complication and is inherently automatic. This system was originally devised by Sir Dugald Clerk for large gas-engines. It consists in admitting above the piston, through ports in the cylinder wall, an additional charge of pure air, or air and cooled exhaust products, after the completion of the normal suction stroke, this supplementary charge being maintained as far as possible in a stratified layer over the piston. It may be utilized either as an inert diluent in order to lower the flame temperature, and so both increase the efficiency of the engine and reduce the heat stresses, or it may be used as an addition to the active working fluid, depending upon whether the initial charge is normal or rich. If rich, then there is sufficient fuel available to saturate the supplementary air; if normal, then the supplementary air acts merely as an inert diluent. Using an ordinary standard carburettor, this gives automatic compensation for altitude, for if the mixture is adjusted to be about normal at ground level, then the supplementary air acts merely as a diluent, while, as the machine rises, and the mixture from the carburettor grows richer, more and more of the supplementary charge becomes active working fluid, so that until a height is attained at which the whole of it is consumed, the torque falls only as the square root of the density and not directly as the density of the atmosphere, while the air speed with a propeller of fixed pitch would remain nearly constant. By employing a cross-head type of piston and making use of the annular displacement of the piston, it becomes possible to add a supplementary charge corresponding to from 30 per cent to 35 per cent of the initial charge, and so to obtain a net increase in torque of about 35 per cent. The general arrangement of an experimental unit built on this principle is shown diagrammatically in fig. 187.

The efficiency of the whole system hinges on the possibility

or otherwise of working with a stratified charge and of obtaining a smooth transition from a stratified to a homogeneous charge.

Fig. 188 shows the calculated indicator card (*a*) when running normally without any supplementary charge and with an economical mixture strength, i.e. 16 : 1 air/petrol ratio, and (*b*) when running with the same mixture but admitting 35 per cent supplementary air as a diluent. Fig. 189 shows the indicator diagrams actually obtained from an experimental engine, under just such conditions, and indicates very fair agreement, the gross fuel consumption being exactly the same in both cases while the torque is increased by approximately 8 per cent. Figs. 190 and 191 are light spring indicator diagrams taken above and below the piston, which show the compression and introduction of the air supercharge. The absence of the "peak" when supercharging is due to the slower burning of the

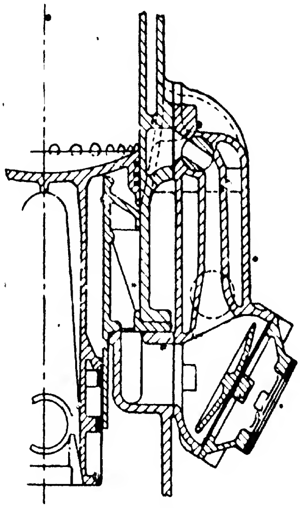


Fig. 187.—Section of lower Part of Cylinder showing Supplementary Air-inlet Ports, &c.

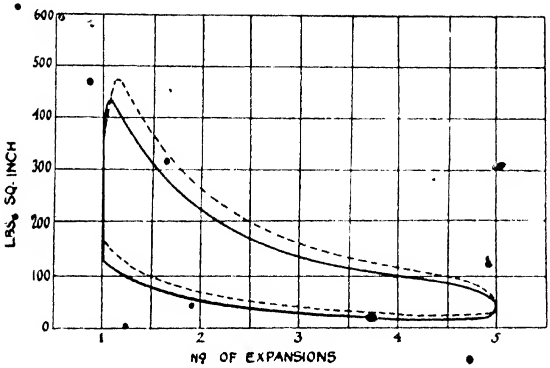


Fig. 188.—Normal Diagram shown in full Lines. Dotted Lines indicate Diagram theoretically obtainable with same Consumption of Fuel, but with Air Content increased by Supercharging.

charge when a large proportion of diluent is present. Actually the total net gain in efficiency, after making allowance for the losses due to pumping in the supplementary air, was found to be

approximately 8 per cent when the whole of the supplementary air was used as a diluent, while the over-all efficiency was almost exactly

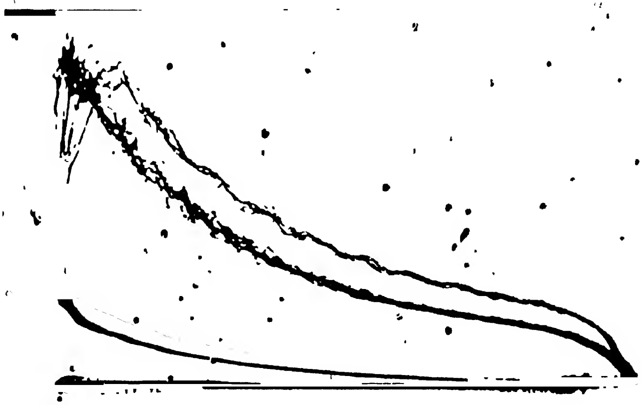


Fig. 189.—Indicator Diagrams Normal and Supercharging

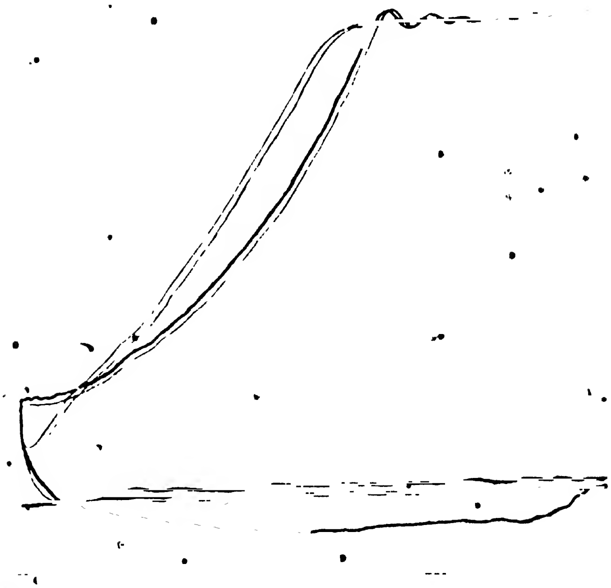


Fig. 190.—Light Spring Diagram from Working Cylinder when Supercharging

the same when the maximum torque was obtained, whether with or without the admission of the supplementary air. In other words, applied to aircraft this method of supercharging by means of a

stratified charge, would give a net increase in fuel economy when the engine is operating at its normal torque of about 8 per cent, or alternatively it would give an increase of about 35 per cent in torque without gain or loss in economy. It would appear to have other advantages also, for, in the first place, it probably eliminates losses due to irregular distribution, since, if any one cylinder receives an over-rich mixture, the result is merely that more of the supplementary air is carburetted, and that particular cylinder develops a greater torque, so that, until a stage is reached when the whole of the supplementary air is used as active working fluid, irregularities in distribution are automatically compensated. Again, this system affords automatic compensation for mixture strength at different altitudes, for, if a normal type of carburettor is used, the torque

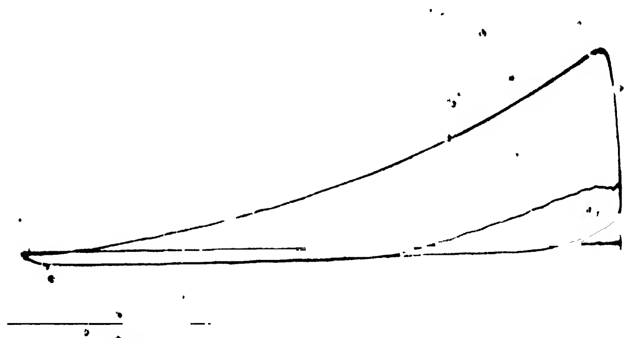


Fig. 191 —Light Spring Diagram Supercharge Chamber

will fall only as the petrol flow, or as the square root of the density, instead of directly as the density as in a normal engine, since, between wide limits, the torque depends not on the density of the surrounding air, but rather upon the flow of fuel.

Unless a variable pitch propeller is used, it is very doubtful whether it is desirable to maintain the torque much higher than that which this system provides.

The curves, fig. 192, give a summary of the performance of a single-cylinder experimental engine operating in the manner described above.

The objections to this system are (1) the comparatively small increase in torque available, when the differential area of the piston is utilized for supercharging, namely about 35 per cent. This, however, can readily be increased by increasing the density of the air supplied to the underside of the piston, in which case, since it

is only the supplementary and not the main air charge which requires boosting, a comparatively small pump or blower will suffice; (2) the additional weight incurred, which amounts to from 10 per cent to 15 per cent.

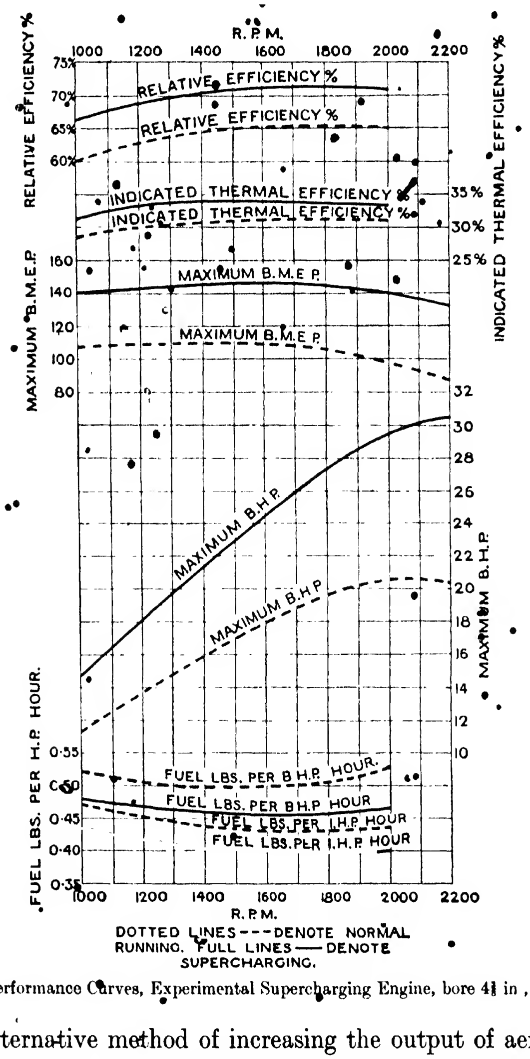


Fig. 192.—Performance Curves, Experimental Supercharging Engine, bore $4\frac{3}{4}$ in., stroke $5\frac{1}{2}$ in.

The alternative method of increasing the output of aero-engines at high altitudes, by increasing the compression-expansion ratio, aims more particularly at an increase in fuel economy rather than in power, for the weight of charge taken into the cylinder per cycle

is not increased, but, on the other hand, more useful work is obtained from a given weight of air, since it is expanded further. This gives an increase both in power output and fuel economy, though the increase in the former is comparatively small as compared with that obtainable by supercharging. There are, however, a good many advantages in connection with this system. With ordinary fuels, the limit of compression and expansion is set by the tendency of the fuel to detonate and ultimately to pre-ignite. This depends mainly upon the chemical constitution of the fuel, but it depends also, as has been explained, upon the maximum flame temperature, the pressure of compression, upon the form of the combustion chamber, and the position of the ignition plug therein. For a fuel of any given chemical constitution the tendency to detonate will become less as the altitude is increased, for both the temperature and pressure of compression will be reduced, as also the maximum flame temperature, which will be reduced in sympathy. It is found that, while the ordinary aviation petrol will tend to detonate at any compression ratio in excess of about 5 : 1 at ground-level density, at about 12,000 feet a compression ratio of 7 : 1 may be used with, at least, equal freedom from detonation. Actual experiments on a variable compression engine have proved that increasing the ratio of compression or expansion from 5 : 1 to 7 : 1 increases the indicated thermal efficiency from 32 per cent to 37·5 per cent, a gain of 16·5 per cent, which corresponds very closely indeed with the theoretical figure predicted by Tizard and Pye. The gain in power is not of the same magnitude because, for some at present inexplicable reason, the volumetric efficiency of an engine falls as the compression is increased. When the compression ratio is raised from 5 : 1 to 7 : 1, the indicated mean pressure was found (in experiments on the variable compression engine) to rise from 141 lb. per square inch to 157 lb. per square inch, a gain of only 12 per cent as compared with the 16·5 per cent gain in economy. Careful measurements of air consumption have proved that the whole of this large discrepancy is to be accounted for by reduced volumetric efficiency, the air consumption per hour for this particular engine at 1500 R.P.M. being 209·5 lb. at a compression ratio of 5 : 1, and 190·0 lb. per hour when the compression ratio is raised to 7 : 1. Though less than might be expected, this gain in torque is by no means to be despised, all the more so since it is obtained without any added complication. When running with a ratio of 7 : 1 the heat stresses are somewhat reduced, and although the maximum pressure on the pistons is

higher, both the pressure and temperature of the gases leaving the exhaust valves are substantially lower—a very important consideration from the point of view of reliability.

The principal difficulty in the way of employing a very high compression engine for high-altitude work lies in operating such an engine at or near ground level. This is so serious a difficulty that, unless some heroic means be adopted, it becomes almost impossible to leave the ground at all. There are several possible ways of attacking the problem. Among these are:

(1) By throttling when at ground level, in order to reduce both the pressure of compression and the maximum flame temperature, the latter because of the greater relative proportion of inert exhaust products to fresh charge, and because of the reduced density generally.

(2) By holding the inlet valve open during a portion of the compression stroke, so that while the expansion is retained, the compression temperature and pressure are reduced.

(3) By adding inert exhaust products, in order both to reduce the maximum flame temperature and the maximum pressure.

(4) By using a special fuel mixture at or near ground level.

With the exception of the last named, all these methods have the disadvantage that they reduce the available power at or near ground level, even when compared with a normal engine having a compression ratio of 5 : 1.

The method of throttling a high compression engine at or near ground level may be dismissed at once as impracticable—not only is it dangerous, but if anything approaching a 7 : 1 compression ratio is used with ordinary aviation petrol, the power output available is not nearly sufficient. The curves, fig. 193, show the maximum indicated mean pressure obtainable with, in this case, a somewhat inferior aviation petrol detonating normally at a compression ratio of 4·85 : 1. The compression ratio was gradually raised and the throttle closed just sufficiently to avoid detonation. It will be seen that at a compression ratio of 7 : 1 the available indicated mean pressure is only 85 lb. per square inch, corresponding in this case to a brake mean pressure of 70 lb. per square inch. This probably would not nearly suffice to raise the machine from the ground.

By the use of variable closing inlet valves in order to vary the compression ratio, somewhat better results can be obtained for various reasons, but even so, the weight of charge is considerably reduced and an extra mechanical complication is added. The

method has, however, some substantial indirect advantages, and, as compared with throttling, it is much safer and yields a somewhat greater power output at or near ground level.

By the addition of cooled exhaust gases, detonation can be suppressed and the maximum pressures reduced. at the least expense in power output of any of the three methods yet considered, but for a compression ratio of 7 : 1 the quantity of exhaust products required is so large that they have an adverse influence on the thermal efficiency as well as on the power output. Also it appears essential

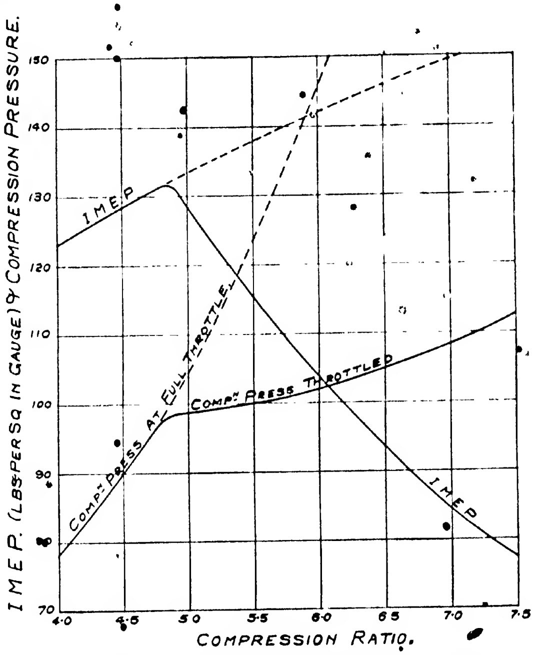


Fig. 193.—Curves showing I M.E.P. and Compression Pressure with varying Compression and Throttling

that they shall be thoroughly cooled before admission to the carburettors or induction system, and this in itself is sometimes rather troublesome. It is, however, in all probability the best of the three methods considered. Fig. 12, Chapter II, shows the results of experiments on the same variable compression engine, when the same fuel was used and the compression gradually raised, just sufficient cooled exhaust products being admitted at each stage to check detonation. It will be seen that by these means the ground-level power output with a compression ratio of 7 : 1 was equal to that

obtained at a ratio of 4·2 : 1, i.e. 125 lb. per square inch i.
or 110 lb. per square inch brake mean pressure. Of the three, this is probably the most hopeful method.

By suitable treatment of the fuel, such as by the addition of toluene, &c., detonation can be eliminated entirely and the full power obtained at ground level, provided that the engine can withstand the excessive pressures involved. At first sight this might appear the simplest and best method, but on investigation it is very doubtful whether it is really practicable, because to withstand the very high maximum pressures involved by the use of a compression ratio of 7 : 1 the whole of the engine, and especially the reciprocating parts, must be strengthened and the weight increased very considerably. The use, however, of a fuel of lower flame temperature and higher latent heat, so that neither the temperature nor pressure is increased appreciably, such, for example, as alcohol, would appear very hopeful.

CHAPTER XII

HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

Although the conditions applying to an engine for tanks are somewhat specialized owing to the peculiar nature of the service required of them, yet, apart from certain features, the following examples may be taken as fairly typical of the class of large high-speed heavy-duty engine developed during the War. Unlike most other heavy transport duties, the engines for tanks were called upon to run for comparatively long periods under very heavy loads, the average load factor when travelling across rough country being over 80 per cent as compared with the 35-45 per cent load factor of ordinary motor lorry engines; again, the engines ran always at their governed speed, which ranged from 1200 to 1350 R.P.M. and averaged about 1250 R.P.M., corresponding to a normal piston speed of 1560 ft. per minute, or about double the average piston speed of motor lorry engines.

Owing to the very large amount of dust and mud imported into the tank by the creeping tracks, the engine was always smothered in dirt or dust, and for this reason it was very desirable totally to enclose the crankcase and to eliminate breathers or any other form of ventilation. Further, they were required to use inferior fuel, and in many cases received only the most scant and unskilled attention.

Owing to the severe gradients which the tanks were capable of negotiating, the engine was frequently required to operate at an angle of over 35° to the horizontal, as shown in the photographs, figs. 194 and 195, which show a tank climbing out of a deep trench, while fig. 196 shows some of the other duties expected of a tank. Further, it was laid down by the authorities that under no circumstances should the engines show smoke from the exhaust. These two conditions necessitated the adoption of special measures both as regards the lubrication and the piston design.

By reason of the low priority under which tanks and their equipment were constructed until the very last phase of the War, only the

cheapest and most easily worked materials could be used. The allowance of aluminium available was so small that it sufficed only for the

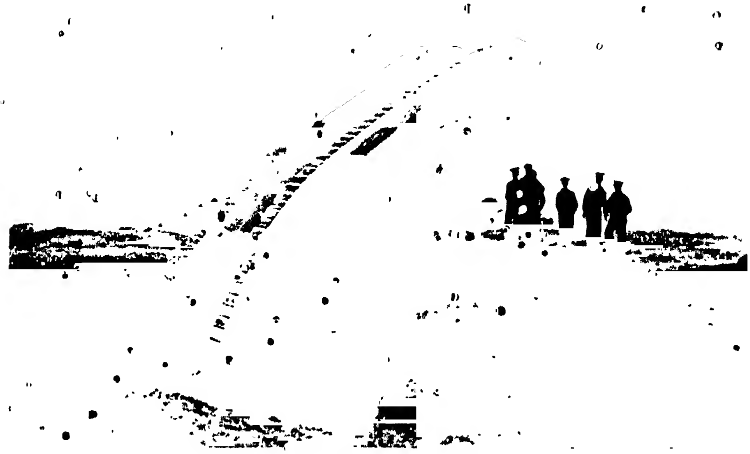


Fig. 194.—Tank at about Maximum Climbing Angle

Fig. 195.—Tank Climbing with Assistance of Unditching Gear

pistons and induction pipes, while the use of high tensile steel was entirely banned.

The standard 150 H.P. type is shown in the photos, figs. 197-200, and 215, and in the drawings, fig. 201. Six separate cylinders

Fig. 196.—Tank crossing wide River

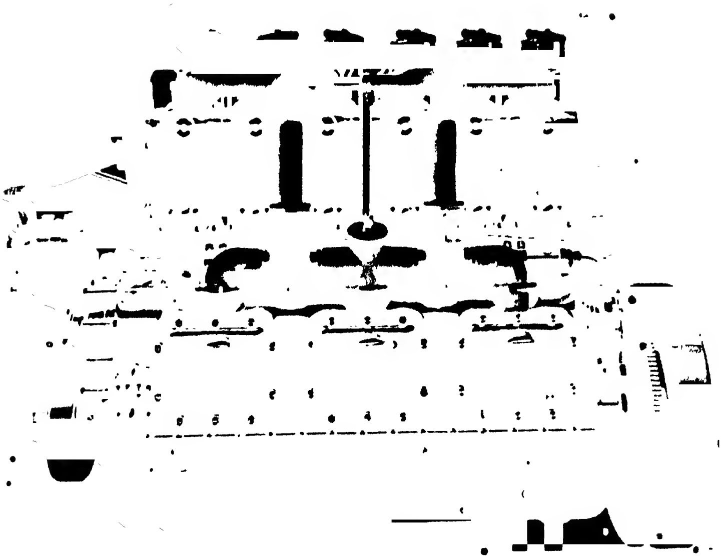


Fig. 197.—150 H.P. Tank Engine, Carburettor Side

are employed, each of $5\frac{5}{8}$ in. bore and $7\frac{1}{2}$ in. stroke; the water jackets are arranged with large openings at the sides, which are covered

with screwed-on sheet steel doors. This form of construction, in addition to facilitating the foundry work, allows of the cylinder centres being brought very close together, thus reducing both the over-all length, which was very limited, and the bending moment on the crankcase, due to the two opposing couples formed by each group of three pistons

The cooling water is delivered to the bottom of the water jacket on the side remote from the valves, and the outlet is arranged between the two sparking plug bosses on the opposite side of the

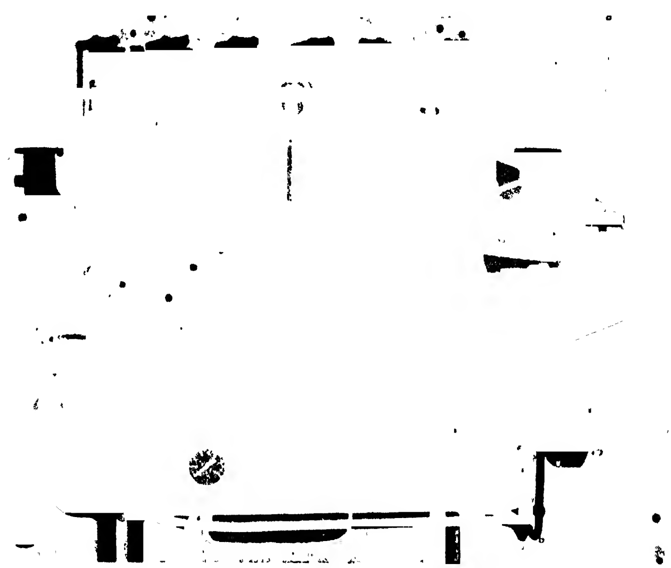


Fig. 198.—150 H P Tank Engine, Exhaust Side

cylinder, the object being to ensure a rapid circulation of water round the sparking plugs.

Provision is made in the cylinder heads for fitting compressed-air starting valves, although this system of starting the engines was never employed.

So far as the exhaust valves are concerned, there is nothing very special to record. Care was taken to ensure the best possible cooling of these by providing a wide seating with an ample supply of water all round, and by using a valve stem of large diameter to conduct the heat away. The valve is cooled by carrying the water as close as possible up to the head of the valve, and also by the use of a valve guide of phosphor bronze, which is an excellent conductor of

heat. There is one feature, however, in connection with the exhaust valves which perhaps calls for comment—that is, they are of 3 per cent nickel steel, case-hardened all over. The object of this treatment was twofold:

(1) Although, of course, the head of the valve does not remain

Fig. 199.—150 H.P. Tank Engine, Flywheel End

hard, the carbonized surface resists pitting, with the result that the seating lasts much longer, and grinding-in is seldom necessary.

(2) The case-hardened stem renders possible the use of a phosphor-bronze valve guide without risk of tearing or seizing.

The connecting-rods are mild-steel stampings of normal design. The only point for comment is in the length of the rods, which are 16 in. between centres, giving an l/r ratio of 4·26 : 1. The principal

reason for the employment of these long rods lies in the fact that it was anticipated that a four-cylinder unit of this engine would be required at a later date, as indeed proved to be the case, and the shorter rods which it would have been possible to employ on a six-

Fig. 200.—150 H.P. Tank Engine, Magneto End

cylinder engine would have been a great disadvantage, in a four-cylinder engine, on account of the secondary disturbing forces.

The top half of the crankcase, or column, is an iron casting of an average thickness of ⅝-in. The general design is clearly shown in the general arrangement drawings, fig. 201, and the function of the false top to the crankcase has already been explained in the chapter dealing with piston design, &c. Inspection doors are fitted

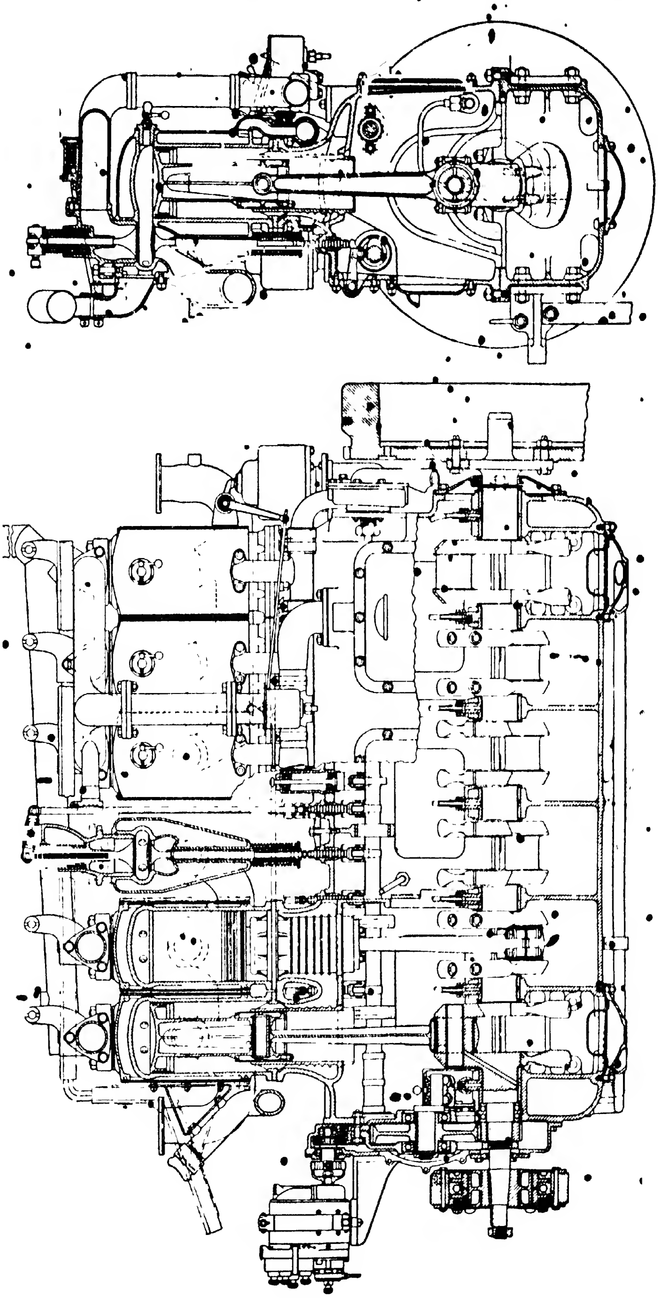


Fig. 201—General Arrangement of 150 H.P. Engine, Longitudinal and Cross Sections

on both sides of the column, and the construction is such that it is possible to remove the connecting-rods, &c., through the inspection doors (fig. 202).

The crankshaft is mounted on seven plain bearings carried in the cast-iron bed-plate; the bearing caps are mild-steel stampings; and the white-metal-lined "brasses" are located in the bearing caps

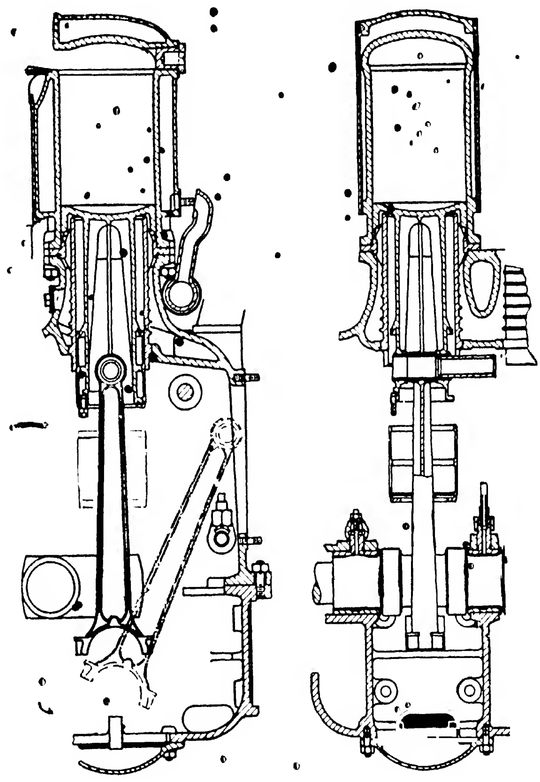


Fig. 202.—Connecting rod, Dismantling Diagram

in order to allow of the removal of both halves of the journal bearings, should this be found necessary, without disturbing either the bed-plate or the crankshaft. Fig. 203 shows the arrangement of the lubrication connections to the journal bearings, and it will be seen that the oil pipe is attached directly to an extension piece cast integrally with the top half of the bearing brass. The extension piece passes through a hole drilled in the steel bearing cap, thus serving to locate the bearing shell. This method of construction

has the advantage that there is less tendency for oil to leak round between the bearing shell and its housing, and so insulate the brasses. The bearings thus dispose of their heat the more readily.

The crankshaft is a mild-steel forging, the principal dimensions of which are given in the table at the end of this description. Owing to the restrictions as regards the length of the engine, the available area of bearing surface was severely limited, and the difficulty of providing adequate bearing areas was still further increased by its being necessary to employ material for the crankshaft of very low surface hardness and having therefore very poor wearing properties. In apportioning the bearing surface between the connecting-rod and journal bearings in the original design, a higher load factor was allowed on the journal bearings and particularly the centre bearing, since this factor could be reduced, if found necessary, by the addition of balance weights.

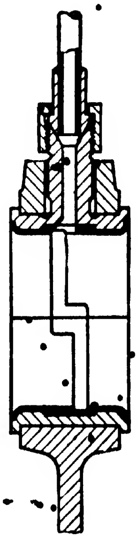


Fig. 203.—Crankshaft Journal Detail

The arrangement for the oil supply to the big ends is orthodox.

The arrangement of expanded-in tube in the crank-pin as shown did not prove altogether satisfactory in service, for it was found that there was a tendency for the annular space to become choked in course of time.

The tube was therefore discarded and replaced by the usual arrangement of two end plugs retained in position by a single through bolt.

The flywheel is an iron casting 26 in. in diameter, and is bolted to a flange formed solid with the crankshaft. A Lanchester vibration damper is attached to the forward end of the crankshaft in order to damp out any torsional vibration.

In order to allow of the engine operating satisfactorily when tilted through large angles, the lubrication is on the "dry base" system; that is to say, the oil supply is not carried in the bedplate, but in a separate oil tank. Three oil pumps of the valveless plunger type are fitted, all three of which are driven from a single crank-pin, which in turn is driven by the intermediate timing gear wheel. The general arrangement of the oil pumps and their driving gear is shown in fig. 204. The centre pump circulates the oil through

the various bearings, and the two scavenge pumps collect a... the used oil to the external oil tank. Each of the scavenge pumps is connected up to one of the small oil sumps which are provided at each end of the bedplate. The lubrication pipework is shown in fig. 205, which also illustrates the oil relief valve at the flywheel end of the main oil lead. In the original design, the scavenge pump suction pipes were arranged externally on the grounds of accessibility of the pipe joints, but in the Mark V Tanks, in which these engines were principally used, the joints were not accessible when the engine

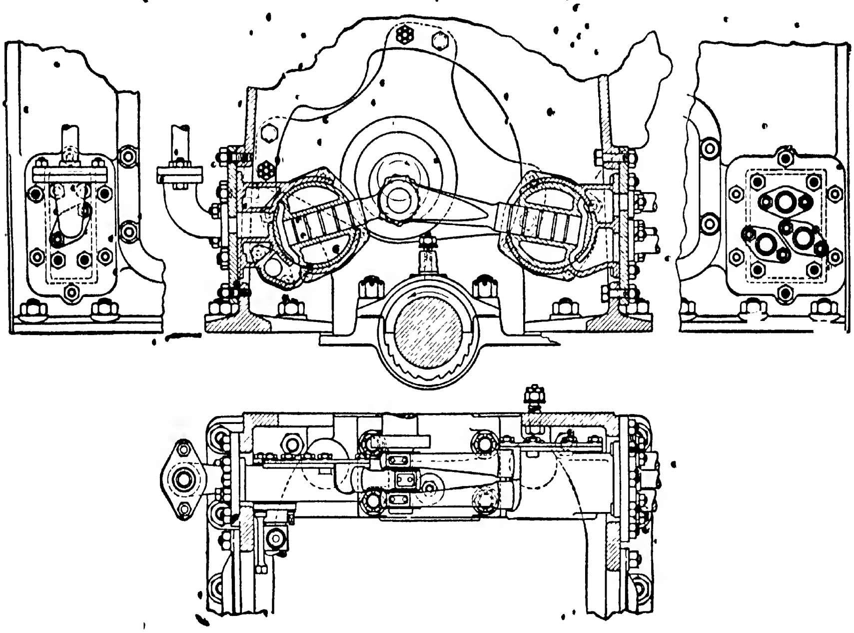


Fig. 204.—Oil Pump Arrangement

was mounted in position. Later engines were therefore fitted with the suction pipes inside the bedplate.

Owing to restriction in width, it was necessary to place all the auxiliaries at the ends of the engine. The auxiliaries to be provided for were as follows: Two magnetos, three oil pumps, two governors, water-circulating pump, and air-pressure pump. The arrangement of the various auxiliary drives will be seen from the illustration, and fig. 206 shows these diagrammatically. In the original design two governors were provided, one to limit the maximum speed of the engine, and the other to open the carburettor throttles directly the

engine speed fell below 400 R.P.M. The object of the second governor was to prevent accidental stoppage of the engine. It was, however, found to be unnecessary, and only the high-speed governor was retained.

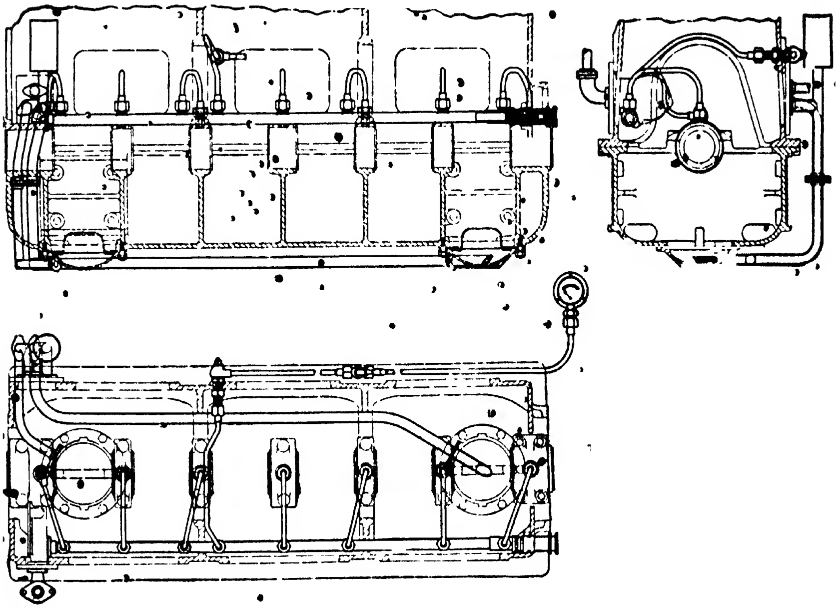


Fig. 205.—Oil Pipe Arrangement

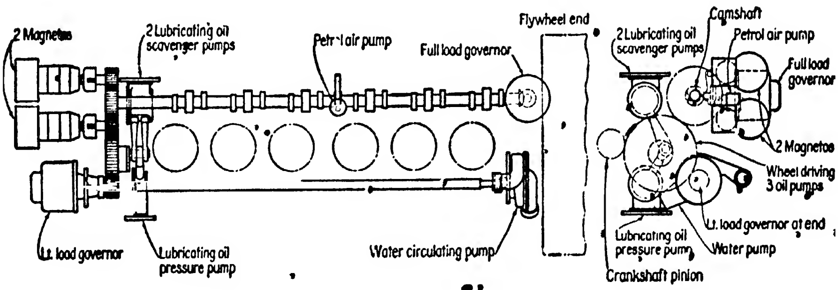


Fig. 206.—Auxiliary Drive Diagram

Fig. 207 shows the arrangement of the water-pump drive. The pumps were designed and made by the Pulsometer Engineering Co., and their performance is shown in the curve, fig. 208.

The intermediate timing gear wheel is mounted on ball bearings carried in a cast-iron spider bolted up to the front wall of the column

a form of construction which facilitates the correct meshing of the timing gear. All three oil pumps are driven by a small disc crank keyed to the hub of the intermediate wheel.

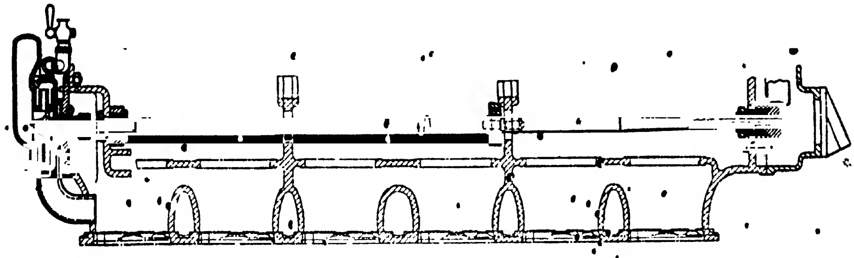


Fig. 207.—Water Pump Drive Arrangement

Fig. 209 shows the general arrangement of the governor; it is a miniature of that used by Messrs. Mirrlees, Bickerton & Day, Ltd., for their large Diesel engines.

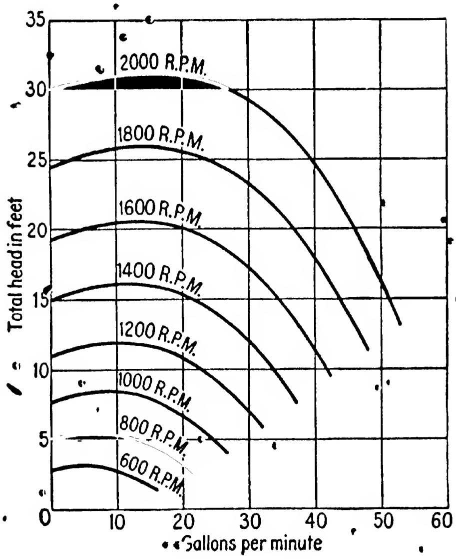


Fig. 208.—Performance Curves of Water circulating Pump

Two 55-mm. vertical Zenith carburettors are fitted, the whole of the air supply to which is taken from the chambers surrounding the cross-head guides; the method of warming the air supply to the carburettors has already been described in connection with the piston

construction. A hand-adjusted cold-air valve is fitted between the two carburettors for use in very hot weather.

All engines were required to pass the following tests before acceptance:

(1) A full-load test of two hours' duration, during which the power

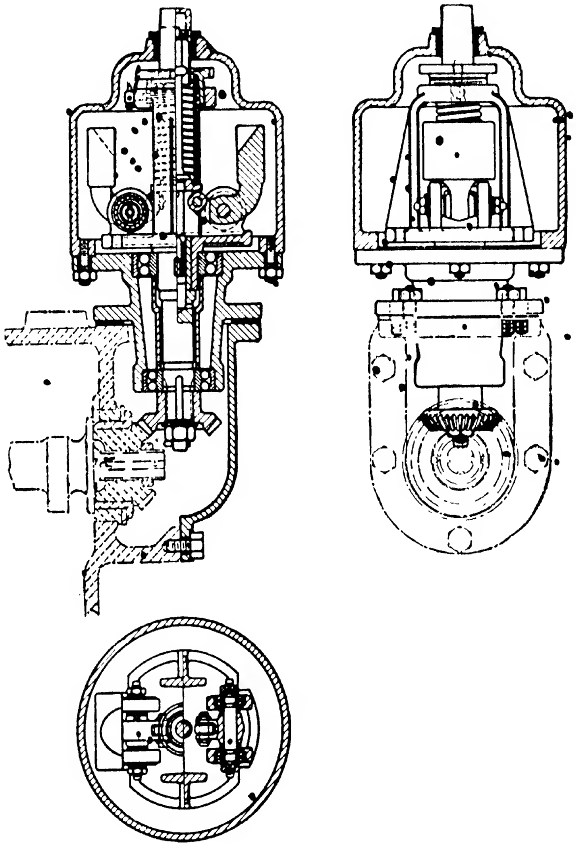


Fig. 209.—Vertical Governor Arrangement

must not fall below 150 B.H.P. at 1200 R.P.M. During this test the fuel and oil consumption was not to exceed 0·7 pint (petrol) and 0·02 pint (oil) per B.H.P. hour.

(2) The above test to be followed by a run of ten minutes at 1600 R.P.M. and not less than 150 B.H.P.

(3) Governor tests.

(4) A low-speed torque test, when each engine was required to develop not less than 55 B.H.P. at 400 R.P.M.

(5) The first engine by each maker and thereafter one in every

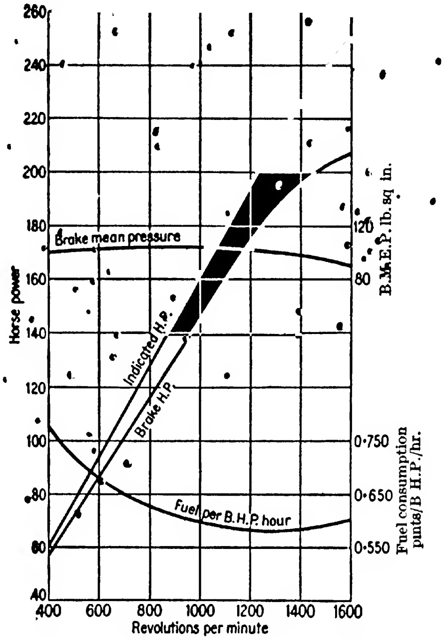


Fig. 210.—Performance Curves, 150 H.P. Engine

fifty, as selected by the Inspector, were submitted to the following additional tests :—

(a) A continuous full-load run of fifty hours, during which the

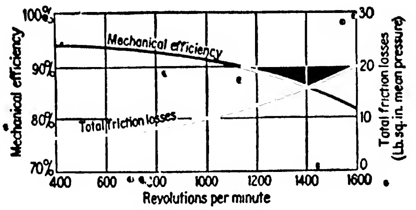


Fig. 211.—Mechanical Efficiency and Friction Losses, 150 H.P. Engir

conditions as to power, fuel and oil consumption were identical with those of the ordinary two-hour full-load test (1).

(b) A tilting test, the engine to be mounted on a tilting table and tilted through an angle of 35° first in one direction and then in the

other. When tilted at this angle the engine was required to be run

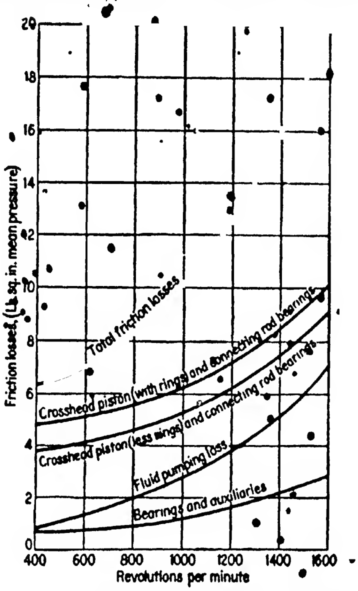


Fig. 212.—Mechanical Loss Curves, 150 H.P. Engine

for ten minutes at about 400 R.P.M. and no load; after this period the throttle was to be thrown wide open, when the engine must open

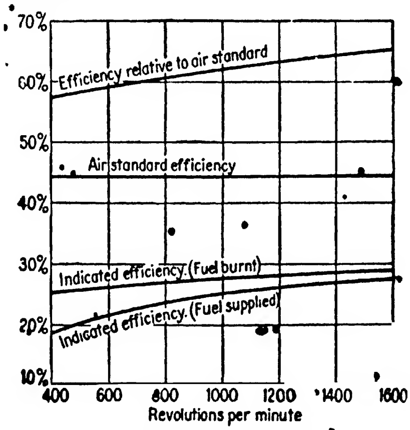


Fig. 213.—Efficiency Curves, 150 H.P. Engine

up firing regularly on all six cylinders without showing any smoke and without any oil leaking out of the base chamber (see fig. 229).

The useful life of a fighting tank was at first so short that an endurance between overhauls of 100 hours was considered ample, but, as might be expected, the engines in actual practice were called upon for an endurance of very much more than the 100 hours originally specified, and at least four instances were reported of engines having run 1400 hours at full speed without requiring or receiving any overhaul beyond the ordinary routine adjustments. Moreover, being of a convenient size and speed, they were used very largely for driving

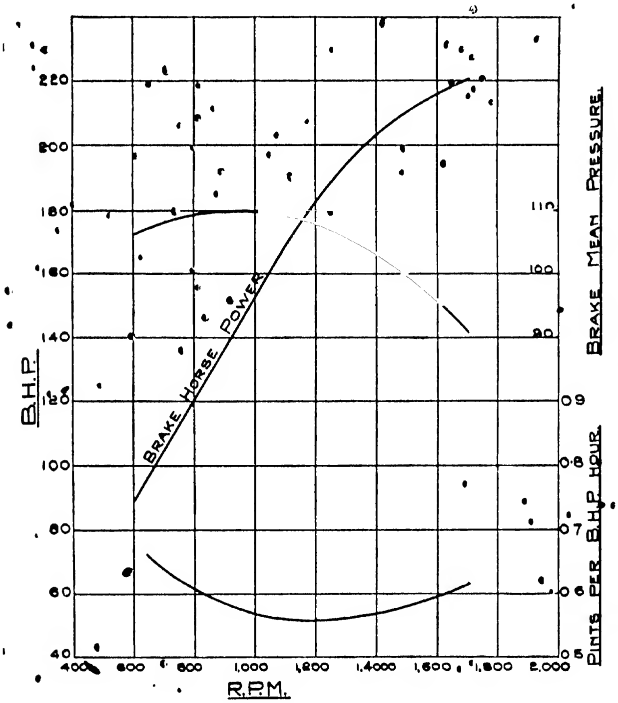


Fig. 214 —Power and Consumption, 150 H.P. Engine

electric generators for supplying light and power to large camps, field workshops, &c., in which service their hours of running were naturally very much longer.

The preceding curves (figs. 210-213) give in full the average performance of these engines. Fig. 210 shows the indicated and brake horse-power, also the brake mean pressure developed at speeds ranging from 400 R.P.M. up to 1600 R.P.M. The brake horse-power and torque curves are the mean of a large number of tests carried out by the different engine-makers, and may be taken as fair average

results. Fig. 214 shows the results obtained from a particularly good example after the conclusion of its 50-hour full-power list. Fig. 212 shows the mechanical losses, which were determined in detail and with considerable accuracy by means of a swinging field dynamometer. All engines on completion of their official full-power run were motored for a short period to determine their mechanical efficiency, and the

Fig. 215.—Sectional Model of 150 H.P. Engine

total mechanical losses were found to agree very closely with the sum of the several detail losses shown in the above curve. Further, in a few instances, tests for mechanical efficiency were carried out by the method employed by Morse, of cutting out one cylinder at a time while the engine is running on full load. These tests also showed very close agreement. All the test sheets show that the mechanical efficiency, as arrived at by the motoring test, was remarkably uniform

over a wide range of engines, a variation of 1 per cent in the mechanical efficiency figure being very exceptional.

Fig. 213 shows the thermal efficiency and the efficiency relative to the air-standard; two efficiency curves are shown—(1) based on the fuel burnt, and (2) based on the fuel supplied. The efficiency based on the fuel supplied is calculated directly from the known fuel con-

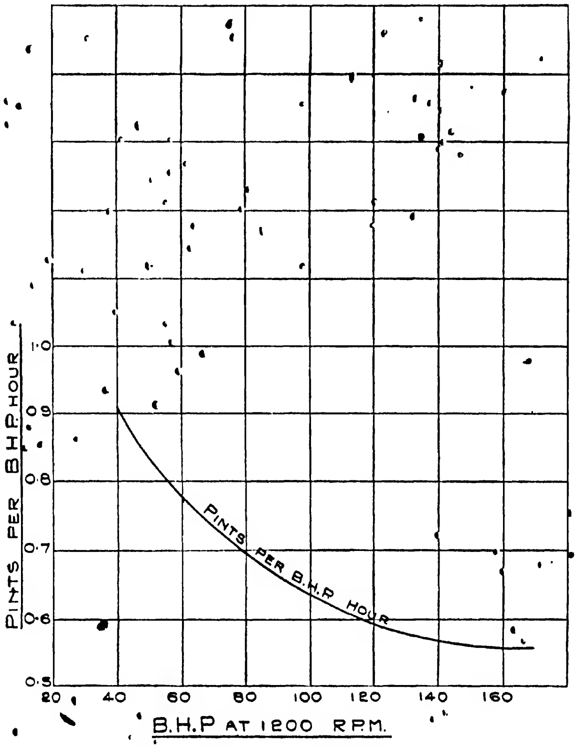


Fig. 216.—Throttle Curve, 150 H.P. Engine

sumption. The efficiency based on the fuel burnt is arrived at by calculating back from the mean pressure actually obtained in the cylinder, and the difference between these two curves represents the loss due to imperfect carburation and distribution.

Fig. 214 shows the power and consumption at full throttle and varying engine speeds.

Fig. 216 shows the fuel consumption at varying loads when running on the governor at speeds ranging between 1200 and 1300 R.P.M.

Fig. 217 shows in detail the cam formation and valve timing.

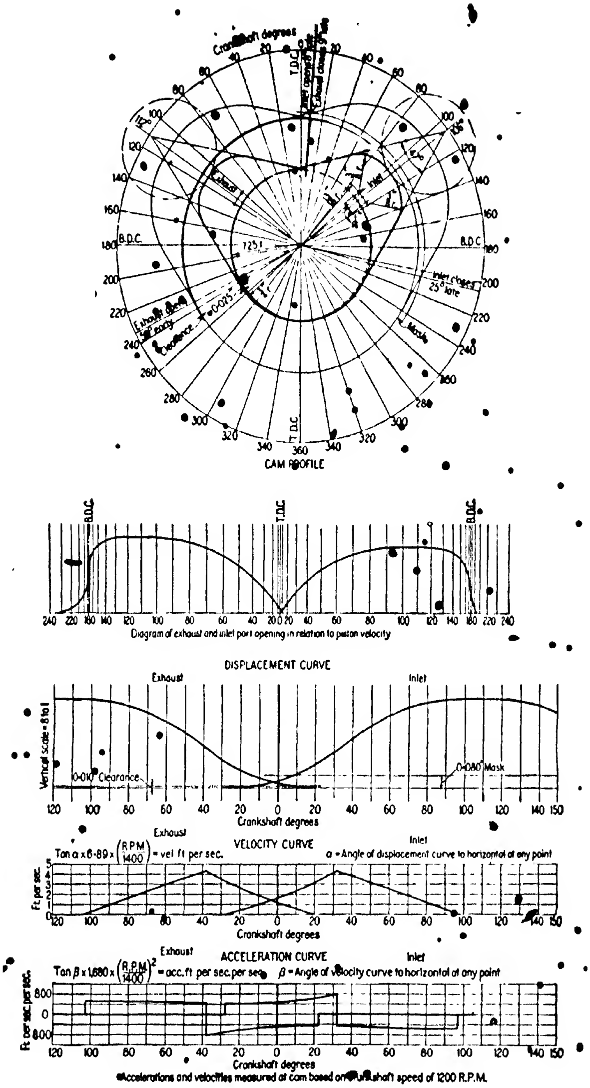


Fig. 217.—Cam Details

The following table taken from a sample test-sheet gives the heat distribution :—

CALIBRATION TEST, 150 H.P. TANK ENGINE

Duration of test.—Ten hours.
Fuel.—Shell spirit (specific gravity 0·725).
Lower heating value of fuel, 18,600 B.Th.U.s per lb.
Air standard efficiency.—44·4 %.
Mechanical efficiency.—87 %.

MEAN RESULTS OF LAST EIGHT HOURS OF TEST.

Brake horse-power	162·9
Fuel (lb./b.h.p.-hour)	0·554
Brake thermal efficiency	24·7 %
Indicated horse-power	187·0
Indicated thermal efficiency	28·4 %
Relative efficiency (per cent of air standard)	64·0 %
Heat loss to jackets (B.Th.U.s per hour)	418,000
Heat to indicated work	28·4 %
Heat to cooling water	24·9 %
Heat to exhaust, radiation, etc.	46·7 %

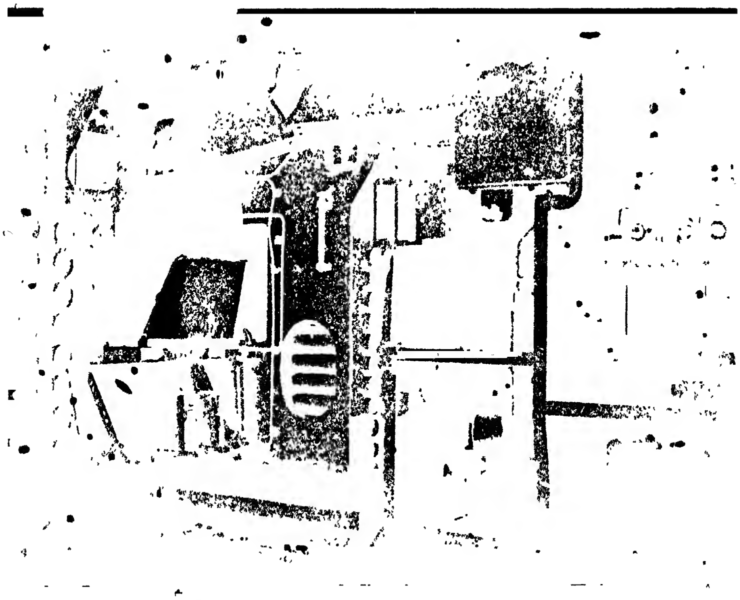


Fig. 218.—Interior of Mark V Two star Tank

Fig. 218 shows the installation of the engine in the Mark V two-star tank, while fig. 219 gives an exterior view. (In this model the larger 225 H.P. engine was used.)

Fig. 219.—No. 106, Mark V Two-star Tank.

The leading dimensions of these engines and general data are given in the following tables :—

Number and arrangement of cylinders ..	Six, vertical, separate.
Bore	5·625 in.
Stroke	7·500 in.
Stroke/bore ratio..........................	1·323 : 1.
Area of one piston	24·85 sq. in.
Total piston area of engine	119·10 sq. in.
Swept volume of one cylinder	186·40 cu. in.
Total swept volume of engine	1118·40 cu. in.
Volume of clearance space	55·90 cu. in.
Compression ratio	4·34 : 1.
Normal b.h.p. and speed	165 b.h.p. at 1200 r.p.m.
Piston speed	1500 ft./min. (25·0 ft./sec.).
Indicated mean pressure	110·6 lb. sq. in. (187·5 i.h.p.).
Mechanical efficiency	88 %.
Brake mean pressure	97·3 lb. sq. in.
Fuel consumption	0·636 pint/b.h.p.hr. (s.g. 0·730). 0·580 lb./b.h.p. hr. (18,600 B.T.U./lb.
Brake thermal efficiency	23·6 %.
Indicated thermal efficiency	26·8 %.
Air standard efficiency	44·40 %.
Relative efficiency	62·0 % (fuel burnt). 60·4 % (fuel supplied).

GAS VELOCITY, VALVE AREAS, ETC.

Gas Velocity (ft. per sec.)—

Choke tube	353·0.

Carburettor body	168·8.
Vertical induction pipe	165·2.
Induction manifold	156·3.
Inlet port	109·6.
Inlet valve	168·3 (c). 130·8 (t.).
Exhaust valve	148·25.
Exhaust port	140·3
Exhaust branch pipes	140·3.
Exhaust manifold	126·6.
Cross Sectional Area (sq. in.)—	
Choke tube	1·760.
Carburettor body	3·680.
Vertical induction pipe	3·760.
Induction manifold	3·976.
Inlet port	5·672.
Inlet valve	3·680 (*c*). 4·750 (*t*).
Exhaust valve	4·130.
Exhaust port	4·430.
Exhaust branch pipes	4·430.
Exhaust manifold	4·910.
Diameter (in.)—	
Choke tube	1·496 in. (38 mm.).
Carburettor body	2·165 in. (55 mm.).
Vertical induction pipe	2·1875 in.
Induction manifold	2·250 in.
Inlet port	2·6875 in.
Exhaust port	2·375 in.
Exhaust branch pipes	2·375 in.
Exhaust manifold	2·500 in.
Weight of piston, complete with rings and gudgeon-pin, etc.	7·25 lb.
Weight per sq. in. piston area	0·292 lb.
Weight of connecting-rod, complete with bearings, etc.	10·70 lb.
Total reciprocating weight per cylinder	10·82 lb.
Weight per sq. in. piston area	0·435 lb.
Length of connecting-rod	16·00 in.
Ratio connecting-rod/crank throw	4·27 : 1.
Inertia pressure, top centre	82·2 lb./sq. in. piston area.
Inertia pressure, bottom centre	51·0 lb./sq. in. piston area
Inertia pressure, mean	33·3 lb./sq. in. piston area.
Weight of rotating mass of connecting-rod	7·13 lb.
Total centrifugal pressure	1094 lb.
Centrifugal pressure, lb. per sq. in. piston area	44·4 lb./sq. in.
Mean average fluid pressure, including compression	43·0 lb./sq. in.
Total loading from all sources, lb./sq. in. piston area	109·5 lb./sq. in.

Diameter of crank-pin	2·875 in.
Rubbing velocity	15·04 ft./sec.
Width of big-end bearing	2·25 in. (effective).
Projected area of big-end bearing	6·47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing	3·84 : 1.
Mean average pressure on big-end bearing	421 lb./sq. in.
Load factor on big-end bearing	6330 lb. ft./sec.
Inlet Valve (*one per cylinder*)—	
Outside diameter	2·875 in.
Port diameter	2·6875 in.
Width of seating	0·0937 in.
Angle of seating	45°.
Radius under valve head	1·9375 in.
Diameter of valve stem	0·4995 in. in guide; 0·4985 in. below guide.
Lift of valve (total)	0·5625 in.
Life of valve (effective)	0·4375 in.
Length of valve guide	4·375 in.
Clearance, valve stem in guide	0·0005 in.
Over-all length of valve	8·375 in.
No. of springs per valve	Two.
Free length of spring	(*a*) 5·875 in. (*b*) 5·8125 in.
Length of spring in position, no lift	(*a*) 4·000 in. (*b*) 3·8125 in.
Mean diameter of coils	(*a*) 1·703 in. (*b*) 1·120 in.
Gauge of wire	(*a*) No. 6 B.W.G., 0·203 in. diam. (*b*) No. 11 B.W.G., 0·120 in. diam.
Valve tappet clearance, cold	0·002 in.
Weight of valve, complete with spring, etc.	1·6234 lb. { Valve bare, 0·8120 lb. Two springs, 0·6718 lb. Spring cap, etc., 0·1406 lb.
Weight of spring, bare	0·6718 lb. per pair.
Inlet valve opens (deg. on crank)	{ Valve leaves seat 29° early. Valve opens 8° late.
Inlet valve closes (deg. on crank)	{ Valve closes 25° late. Valve back on seat 59° late.
Material for valve	3% nickel steel stamping, case-hardened.
Exhaust Valve (*one per cylinder*)—	
Outside diameter	2·6875 in.
Port diameter	2·375 in.
Width of seating	0·156 in.
Angle of seating	45°.
Radius under valve head	0·875 in.
Diameter of valve stem	0·5595 in. in guide; 0·5475 in. below guide.
Lift of valve	0·5625 in.
Length of valve guide	7·125 in. (effective).
Clearance, valve stem in guide	0·003 in.

Over-all length of valve	12·8125 in.
No. of springs per valve	Two.
Free length of spring	(a) 5·875 in. (b) 5·8125 in.
Length of spring in position, no lift	(a) 4·000 in. (b) 3·8125 in.
Mean diameter of coils	(a) 1·703 in. (b) 1·120 in.
Gauge of wire	(a) No. 6 B.W.G., 0·203 in. diam. (b) No. 11 B.W.G., 0·120 in. diam.
Valve tappet clearance, cold	0·010 in.
Weight of valve, complete with spring, etc.	2·0934 lb. { Valve bare, 1·281 lb. Two springs, 0·6718 lb. Spring cap, etc., 0·1406 lb.
Weight of spring, bare	0·6718 per pair.
Exhaust valve opens, degrees on crank	53° early.
Exhaust valve closes, degrees on crank	9° late.
Material for valve	3% nickel steel stamping, case-hardened.

Connecting-rod—

Length between centres	16·00 in.
Ratio connecting-rod/crank throw	4·27 : 1.
Little-end bearing, type	Chilled phosphor-bronze bush fixed in rod.
Little-end bearing, diameter	1·250 in.
Little-end bearing, length	1·625 in.
Little-end bearing, projected area	2·032 sq. in.
Big-end bearing, type	Bronze shell, lined white metal.
Big-end bearing, diameter	2·875 in.
Big-end bearing, length	2·6875 in.
Big-end bearing, projected area	6·47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing	3·84 : 1.
Number of big-end bolts	Four.
Full diameter	0·500 in.
Diameter at bottom of threads	0·4375 in.
Total cross-sectional area at bottom of threads	0·602 sq. in.
Total load on bolts, at 1200 r.p.m.	2659 lb.
Total load on bolts, at 1600 r.p.m.	4795 lb.
Stress per sq. in., at 1200 r.p.m.	4490 lb./sq. in.
Stress per sq. in., at 1600 r.p.m.	7965 lb./sq. in.

Crankshaft—

Length of complete shaft	66·00 in.
Cylinder centres	7·250 in.
Cylinder centres (centre pair)	8·000 in.
Outside diameter of crank-pin	2·875 in.
Inside diameter of crank-pin	1·4375 in.
Length of crank-pin	2·750 in.
Outside diameter of journals	2·875 in.
Inside diameter of journals	Solid.
Number of journal bearings	Seven.

Length, flywheel-end bearing	1·000 in.
Length, forward-end bearing	2·875 in.
Length, centre bearing	2·875 in.
Length, intermediate bearing	2·125 in.
Width of crank webs	3·500 in.
Thickness of crank webs	1·1875 in.
Radius at ends of journal and crank-pins	0·250 in.
Diameter of drilled oil-ways in shaft	0·250 in.
Weight of complete shaft	182 lb., includes thrust race pinion, starting clutch, nuts, etc. (no balance weights).
Material	Mild-steel forging.

It will be interesting and perhaps helpful to review the defects which revealed themselves in these engines. With over 4000 engines in service it is possible to discriminate between mere accidents and epidemic troubles.

Combustion Chamber.—The design of the combustion chamber, though excellent from the point of view of turbulence, and therefore of power output and efficiency, proved, as in the light of present knowledge might have been expected, rather bad from the point of view of detonation, despite the comparatively low compression ratio. Originally it was intended to operate the tanks only on aero-spirit, but later they were required to use the lowest grade of war spirit—an American fuel consisting almost entirely of the heavier fractions of the paraffin series. With this fuel detonation became severe when the engines were pulled down to a low speed with wide-open throttle.

Pistons.—The first few engines were fitted with sand-cast pistons in 88 per cent aluminium and 12 per cent copper alloy. These castings sometimes gave trouble owing to porosity of the metal at the point where the hollow trunk joins the head of the piston. Many pistons were rejected on this score, but a number in which the porosity did not appear on the surface and therefore was not detected by inspection were fitted to engines, and some of these broke away at this point, but since the connecting-rod was not released by such failures little or no further damage resulted therefrom. This defect was remedied completely by employing pistons cast in metal dies in all subsequent engines.

Cross-head Guides.—These were at first made of bronze lined with white metal, and proved quite satisfactory. Owing to the scarcity, or alleged scarcity, of bronze, the use of this material was eventually forbidden by the authorities, and cast iron was therefore

substituted. The cast-iron guides gave a good deal of trouble owing to distortion after machining, and to meet this, since there was no time for "ageing" or annealing, it became necessary to allow rather a large working clearance, which gave rise to noise. Later the same copper-aluminium alloy used for the pistons was employed also for the guides, and this proved extremely satisfactory—quite equal to the white-metal-lined guides as regards wear, and better in so far as, since the clearance increased with temperature, it was safe to work with a very close fit.

Exhaust Pipes.—The exhaust manifolds and pipes radiated so much heat that it was found necessary, for the comfort of the tank crew, to jacket them with an air jacket through which a circulation of cold air was induced by means of a fan driven from the engine.

Crankshafts.—Owing in part to the small bearing area necessitated by the limits on the over-all length of the engine, and in part to the use of dead soft steel, the wear in the journals, and in particular the centre journal, was rather severe. So far as the actual fighting tanks were concerned, this was of little consequence, because the crankshaft easily outlasted the rest of the machine; but in the case of the tanks used for training, which ran all day and every day, and of electric-generating sets, this wear became troublesome, for it was generally necessary to regrind the crankshaft and fit new bearings after from 500 to 700 hours' running at full speed and 80 per cent load factor. It was therefore decided to fit balance weights on all engines destined for other than fighting tanks, a procedure which reduced the rate of wear to less than one-third, but which, by lowering the natural periodic speed of the shaft and by increasing the intensity of the oscillations, threw a heavy duty on the torsional vibration damper.

Apart from wear, the soft mild-steel crankshafts were very reliable indeed, and, out of over 4000 engines, no single instance of broken crankshaft was reported.

Vibration Dampers.—These were at first identical with those fitted to the Daimler engines. So long as the crankshafts were unbalanced, the duty on these dampers was very light indeed, and they gave no trouble. But so soon as balance weights were fitted and the dampers were called upon to function, the thin plates used in them soon cut away through the castellations and the damper wore out completely in about 50 hours' running. As the result of this experience the dampers were re-designed by Messrs. Gardner & Sons; two thick cast-iron plates with good substantial castellated bosses

being fitted in place of the rest of thin steel plates). these proved perfectly satisfactory, the wear, even after 1200 hours' running with balance weights, being reported as almost negligible.

Lubrication System.—As explained previously—two scavenging pumps exhausted the oil, one from either end of the necessarily very shallow crankcase, and delivered it to a filter tank placed some 3 feet above the level of the crankshaft. A third force pump drew oil from the filter tank and delivered it to all the crankshaft bearings. All three pumps were operated from a single crank at the forward end of the engine, and had the same stroke; but the scavenging pumps were of slightly larger bore, and so could exhaust more rapidly than the force pump could deliver. It was found, however, that under certain conditions when the forward part of the engine was tilted up at the extreme angle the suction pump drawing from the after end failed to keep pace, with the result that oil gradually accumulated in the after end of the crankcase till it eventually ran out of the flywheel bearing. This was found to be due to the fact that while the force pump had a positive head of about 3 feet in its favour the scavenge pump had, under extreme conditions, a suction head of about 3 feet, while the length of suction pipe of about 5 feet was subject to a pulsating flow. Under these conditions the volumetric efficiency of the force pump exceeded that of the scavenge pump by more than the difference in volumetric capacity. This difficulty was cured completely by the provision of an air vessel on the suction side of the scavenge pump, thus maintaining a uniform flow in the suction pipe.

Again, it was found that, when the engine stopped in a certain position, oil could leak back from the filter tank through the bearings into the crankcase, sufficient in time to flood the latter. This occurred only when the machine was left standing for several days in very hot weather. To obviate this defect a cock was fitted in the oil pipe leading to the force pump, and, in order to render it impossible to start the engine with the oil supply cut off, this cock was combined with the magneto earthing switch.

Apart from these two minor difficulties, which were easily overcome, the lubrication system worked well; and although the scavenge pumps were considerably above the highest oil level, and under normal conditions their suction inlets also were above, yet they never failed to pick up the oil at once so soon as the engine was tilted and one or other of the suction inlets was drowned.

Valve Spring Caps.—In order to economize machining opera-

tions the valve spring caps were pressed out from sheet steel, and with a view to reinforcing the conical portion and prevent any risk of its being drawn over the taper wedges, a steel wire ring was embodied in the pressing, as shown in fig. 220. In practice these pressed spring caps used to fail after a period ranging from 200 to 600 hours, the failure invariably occurring at the junction between the conical portion and the flat retaining face, as shown. The failure was due in part to fatigue and in part to the fact that, in spite of all precautions, the pressing was generally sharply nicked at this point. Though no doubt with a little more care in manufacture satisfactory pressings could have been made, it was decided, in view of the urgency, to employ spring caps turned from steel bar, which proved quite satisfactory.

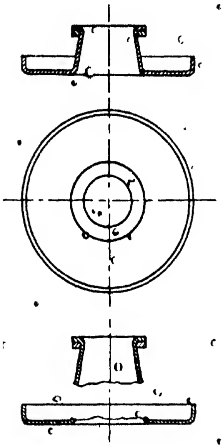


Fig. 220

Valve Gear.—With a view to eliminating noise and wear in the valve gear the base circle of the cams was ground slightly eccentric in order to permit of a large working clearance at the time when it was essential that the valves should be closed, and gradually to take up this clearance preparatory to opening. This method is very satisfactory as a rule, but it is liable to abuse. In practice it was at first found very difficult to prevent the mechanics in charge from adjusting the clearance regardless of the position of the cam, with the result that the valves were lifted slightly from their seats long before the correct opening period. So far as the inlet valves were concerned no trouble arose, but a few cases of burnt-out exhaust valves were found to be due to this cause. The difficulty was eventually overcome by the issue of very full instructions for the adjustment of valve clearances and by rigorous inspection. Once the correct adjustment was fully understood, trouble with valves became almost unknown, and there is little doubt but that the immunity from trouble and the quiet running obtained were largely due to the use of cams with eccentric base circles. Apart from these troubles, of which all but the tendency to detonate and the heat radiated from the exhaust were quickly and easily overcome, these engines behaved very well. The system of hermetically sealing the crank case and drawing all the air through the false top proved most effective in keeping the crankcase cool and the working parts clean.

The cross-head type pistons proved thoroughly effective both in eliminating the usual troubles with large aluminium pistons, in preventing smoke, and in yielding a very high mechanical efficiency.

With the steady growth in size and armament of tanks it became necessary to provide still larger engines, and another six-cylinder engine of the same general type but embodying all the experience gained with the 150 H.P. engine was next designed and produced. This engine is shown in the drawings and photos: figs. 221, 222, 223, 224, 225. Although about 800 of these engines were completed, yet owing to delays in tank production none of them saw any active fighting. Sufficient experience was, however, gained both

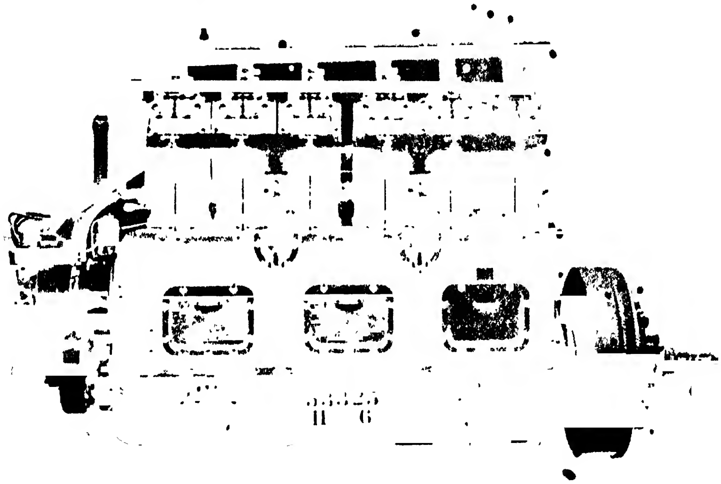


Fig 221.—225 H.P. Engine, Carburettor Side

on the test-bed and subsequently in tanks to prove that the larger engines were a very great improvement over the earlier type.

Rated normally at 225 B.H.P., they had a bore and stroke of $6\frac{3}{4}''$ and $7\frac{1}{2}''$ respectively and developed 260 B.H.P. at 1200 R.P.M.

In order to obviate the defects of the 150 H.P. engines the following modifications were made:—

The combustion chamber was made in the form of a compact rectangular chamber with the ignition plugs placed as nearly as possible in the centre, as shown in fig. 225. This resulted in complete immunity from detonation under all conditions, so much so that the engines could be loaded down to two or three hundred R.P.M. with wide-open throttle without the least trace of detonation even

on the lowest grade of American petrol, and this despite the fact that the cylinders were larger and the compression ratio as high as in the 150 H.P. engines.

The valves were duplicated and placed horizontally with the inlets on one side and the exhaust on the other. This arrangement

Fig 222.—225 H.P. Engine, Flywheel End

of valves permitted of the exhaust ports being turned up vertically so that short straight exhaust pipes could be taken direct through the roof of the tank to a manifold fitted outside. Thus the length of exhaust pipe inside the tank was reduced to the very minimum, while all bends, &c., were avoided. The inlet valves were fitted in separate cages which could easily be removed without disturbing any of the pipe work, and the exhaust valves could then be

Fig. 223.—225 H.P. Tank Engines in course of erection at Messrs. Gardner's Works

withdrawn through the openings left by the removal of the inlet valves.

Die-cast pistons were used from the very start, and no single instance of piston failure was ever reported. There being no rigid restrictions as regards length, more liberal bearing surfaces could

Fig. 224.—225 H.P. Engine, mounted for Tilting Test

be and were provided, particularly as regards the crankshaft centre bearing. Balance weights and Messrs. Gardner's revised design of torsional vibration damper were fitted in all cases.

The crankshafts were made from 0·35 carbon steel, and under these circumstances proved practically immune from wear. The

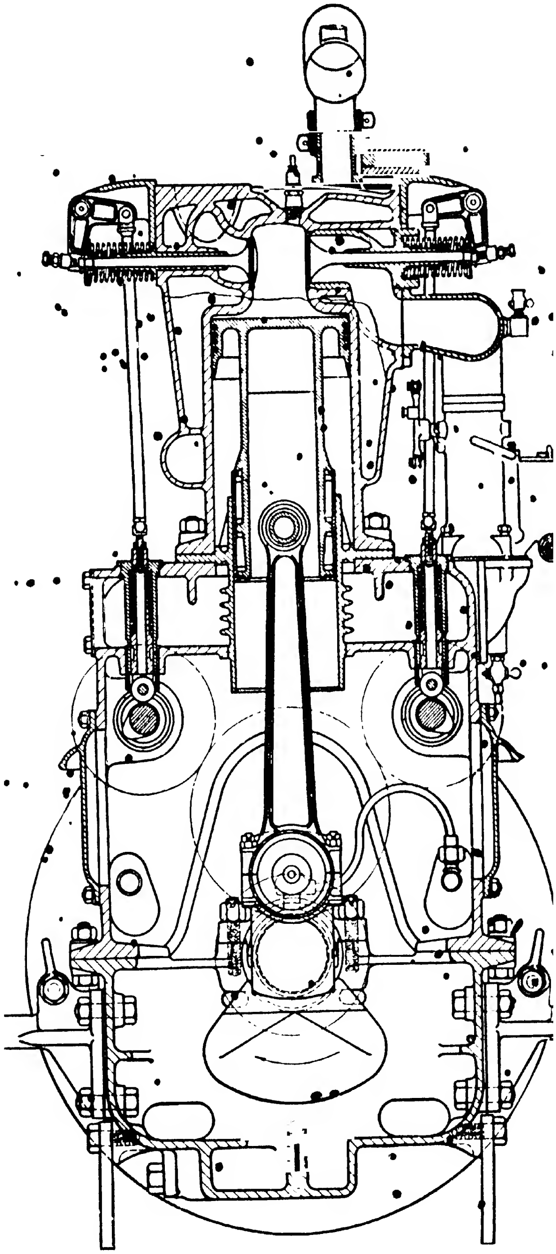


Fig. 225.—225 H.P. Tank Engine, Sectional Elevation

other defects, which manifested themselves in the first batches of 150 H.P. engines were, of course, obviated by adopting the expedients explained above.

Although these larger engines never saw actual fighting service, yet they were submitted to very severe tests both in tanks and on the test-bed, the latter including full-power runs of 200 hours' duration. As may be supposed, with the experience of the smaller engines available before the design was commenced, these engines were an improvement on the 150 H.P. type in almost every respect. A very large number of them have since been converted to run on gas, and are now in use in electric-power stations in various parts of the country. Unfortunately, however, their compression ratio is too low for efficient running on town or producer gas.

The author desires to thank the proprietors of "The Automobile Engineer," and Messrs. L. Gardner & Sons for permission to use several of the illustrations in this volume.

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